

Treball de Fi de Grau

## **Grau d'Enginyeria en Tecnologies Industrials**

**Design analysis, study, set-up and improvement of the fore  
and aft suspension system of the Dallara F317**

### **MEMÒRIA**

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## Summary

The present project covers the design study, analysis and geometric set-up of the fore and aft suspension system of the single-seater racing car Dallara F-317.

The goal is to determinate a geometric configuration of the suspension system that improves the base setup determined by the manufacturer for a specific racing track, that in the case of this project is Spa-Francorchamps. The modelling and virtual simulation tools that are used in the project allow to achieve the defined goal.

The project shows a brief introduction to the history of the Formula 3 category, the technical specifications of the Dallara F-317 and its suspension system. Using the software Optimum kinematics, the modeling, analysis and simulation of the base configuration of the mentioned car is defined. New configurations of the suspension system are defined, simulated and compared to the base configuration in order to test their performance according to the standards established. Later, by using the software Optimum Lap, the performance of the racing car with the standard configuration and the proposed one is measured in a specific track in order to validate if the proposed solution offers an improvement for that given track. Inally, a new geometric configuration of the fore and aft suspension system of the Dallara F-317 is chosen as the better option for the track above mentioned.

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# 1. Preface

After several years studying in the ETSEIB school and being part of the Formula Student team for 3 consecutive years I noticed the amount of technical knowledge and skills I had acquired overtime. However, speaking to people the general impression I had was that the vast majority believes all that knowledge is specific to the project is used with, and that it has no relevance or application outside in a 'real world' scenario. I had the possibility to work for a year in SEAT's Technical Centre in Martorell making different studies and analyses related to the suspension system of production and prototype cars where I could prove that the knowledge acquired in the Formula Student team was useful to me.

That is the main reason behind the making of this project: to show, that all that knowledge can be applied to set-up a real-world racing car using those same techniques learned in the Formula Student team and make it faster lapping around a track.

## 2. Introduction

Setting up a racing car is a crucial step in order to achieve the best possible result for any racing team and a capital requirement in order to have any chances of achieving victory. Therefore, understanding each of the vehicle's systems and the changes in the car's behaviour due to each of the parts' modification is very important to maximize the performance of the car in different tracks.

Single-chassis, single-engine and single-tyre racing series accentuate the above-mentioned aspects even more, as the competitors build upon the same material and the correct set-up can be the determinant factor.

The project's aim is to reproduce some aspects of the aforementioned scenario; setting up a specific part of the car in order to maximize its performance in a given track.

Out of all the available systems to modify in a racing car, the suspension system plays a key role in maximizing the contact of the tyre with the ground and managing the driveability of the car. Therefore, it is the reason why said system is chosen for the study.

The diagram below shows the different available systems for setting up a racing vehicle and the one chosen for this study.



Figure 2.1: Diagram of the different systems to set up a racing car.

## 2.1. Subject

The project is focused around the analysis of the kinematics of the suspension system of the Dallara F-317 single-seater and its impact on race track lap times, while maintaining the rest of the components of the car fixed, such as the aerodynamics, the engine and the wheels. The configuration used for these elements is the one specified in the championship rules and in the manufacturer's technical specification sheet.

## 2.2. Specific objectives

To achieve the general aim of the project, the following specific objectives are defined:

1. *State of the art general overview:* Overview of the single-make racing series and championship format, cars competing in the series and the manufacturer that produces the Dallara F-317. Technical specifications of the car subject to study.
2. *Analysis of the Dallara F-317 racing car:* Vehicle technical specifications, suspension system geometry disposition and overview.
3. *Theoretical introduction to a vehicle's suspension system:* General overview of suspension types, differences between a road car and a racing car and types of racing car's suspension systems.
4. *Dallara F-317 suspension system configuration analysis:* geometry generation, definition of the base configuration and kinematic analysis.
5. *Optimization of the Dallara F-317's suspension system configuration:* modification of the different setup parameters, elaboration of a new configuration and comparison with the base configuration.
6. *Lap time simulation of the Dallara F-317:* Simulation on a virtual track and comparison of the base and proposed configuration.

State of the art	Dallara F-317	Suspension systems introduction	Dallara F-317's suspension system configuration	Optimization of the suspension system's configuration	Lap time simulation
<ul style="list-style-type: none"> <li>• F3 overview</li> <li>• Championship format</li> <li>• Manufacturer</li> </ul>	<ul style="list-style-type: none"> <li>• Technical specifications</li> </ul>	<ul style="list-style-type: none"> <li>• Overview</li> <li>• Racing car vs Road Car</li> <li>• Suspension system types</li> </ul>	<ul style="list-style-type: none"> <li>• Geometry generation</li> <li>• Kinematic analysis</li> </ul>	<ul style="list-style-type: none"> <li>• Iterarions</li> <li>• Tuning of the parameters</li> <li>• Comparison</li> </ul>	<ul style="list-style-type: none"> <li>• Comparison</li> </ul>

Figure 2.2: Summary of the objectives treated in the study

## 2.3. Scope

This project aims to reproduce similar steps as a racing team does when setting up the geometry and kinematics of the suspension system of a race car, except for the real verification and correlation of the data in a real track test day due to the impossibility of working with a racing team during the making of this project and the impossibility of purchasing the vehicle for the study.

Therefore, the project covers the following phases:

1. Design analysis of the Dallara F-317's suspension system. Here an overview of the characteristics of the configuration of the suspension system is presented along with its kinematic parameters.
2. Kinematic study of the base configuration. Utilizing Optimum Kinematics, different kinematic parameters of the suspension are analysed and studied in order to determine its relevance in the behaviour of the car during different types of motions.
3. Optimization of the base configuration. Here different configurations are tested with the aim of improving the performance of the car in the chosen track. Graphics and tables are used to show the comparison.
4. Lap time simulation of the Dallara F-317. Utilizing Optimum Lap, the base and the chosen configuration are tested in a virtual representation of the track in order to quantify the differences between configurations with lap times.

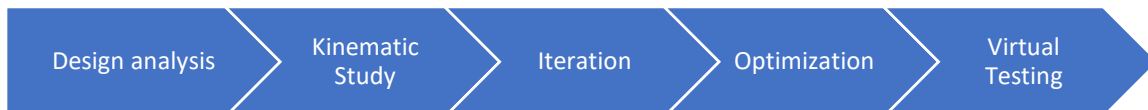


Figure 2.3: Phases of the study of this project



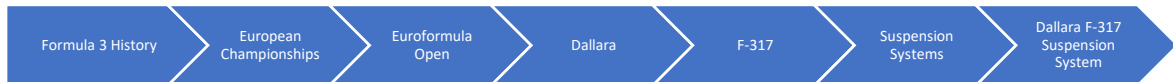
## 2.4. Design Requirements

The following list contains the set of concepts and software recommendable for the successful realization of this project.

1. *Motorsports knowledge:* Knowing what open-wheel, single-seater racing cars are, single-make championships and its regulations.
2. *Vehicle dynamics basic knowledge:* Especially focused around the suspension system and its geometric variables. It is needed to determine the movements of the car and how the suspension system reacts and changes according to those, as well as for determining and analysing all the kinematic and geometric parameters of it.
3. *Microsoft Office:* Software used to accomplish tasks such as programming the masterplan of the project, writing the information of the document, displaying the graphics and preparing the presentation.
4. *Optimum Kinematics:* Vehicle kinematics software used to analyse, modify and calculate results of the different suspension kinematic parameters.
5. *Optimum Dynamics / Optimum Lap:* Vehicle dynamics software used to virtually analyse a vehicle in a given track in order to quantify the performance on track of the different systems of said vehicle.

### 3. Memory's Core

This section covers the origins of the vehicle of the study, the design analysis, the study of its baseline suspension system configuration, the creation of a new configuration and the comparison between the two. The beginning is an overview of the history of the category until the ground level of study is reached with the analysis of the Dallara F-317's suspension system configuration.



*Figure 3.1: Sequence of the different topics discussed in this section*

#### 3.1. State of the art

This subtopic covers a brief overview of the origins of the competition and the car's manufacturer, Dallara Automobili

##### 3.1.1. The Formula 3

Formula 3 is considered as one of the most important junior categories in international single-seater racing. For more than five decades it has been the ideal training ground for the future stars of motor racing. With its numerous European, American, Asian and Australian championships, it is considered to be the best school for professional drivers.

In the years after World War II, first efforts were made to establish motor racing with small single-seaters, often powered by motorbike engines. The goal was to create the possibility to enter motor racing at low costs. But only in 1957, an Italian racing driver and journalist called Giovanni Lurani succeeded in establishing a generally accepted category: the Formula Junior. It is still regarded as the beginning of Formula 3 nowadays. In 1964, Formula Junior was replaced by a new 1.000 cc single-seater category that was named Formula 3.

Before long, this category proved to be successful all over Europe, with renowned racing car constructors such as: Lotus, Brabham and Cooper. In many countries, national championships were launched. In 1975, an official Formula 3 European Championship was established for the first time that would exist until 1984.

Four-time F1 World Champion Alain Prost, who won the 1979 F3 European Championship was arguably the most famous driver who learned his trade in this series before advancing to F1. Riccardo Patrese (1976) and Michele Alboreto (1980) also made it to the pinnacle of motor racing after having won the European championship. [1]

In addition, there were numerous national F3 championships that prospered for many years. Here, the likes of the future Formula 1 World Champions Jackie Stewart, Emerson Fittipaldi, Ayrton Senna, Mika Häkkinen and Jenson Button took part in F3 championships. While the most renowned German F3 Champion was future F1 record champion Michael Schumacher who won the German F3 title in 1990.

A higher-ranking series was founded in 2003: The Formula 3 Euro Series. On the one hand it continues the idea of a European Formula 3 championship and on the other hand because of the merger of the French and German national F3 championships it became the successor of these two-race series. Besides the Euro Series and the German Formula 3 Cup there are further national series in other countries worldwide, e.g. Spanish, Italian and British Formula 3. Japanese Formula 3 is considered the strongest Formula 3 series outside of Europe.

The most famous young talents that advanced from here to Formula 1 were later World Champions Lewis Hamilton, Sebastian Vettel and Nico Rosberg. In 2011, the 'FIA Formula 3 International Trophy' was created and in the following year, this series was replaced by the 'FIA European Formula 3 Championship'. Its race calendar comprised Formula 3 Euro Series and British Formula 3 Championship rounds.

Since 2013, the 'FIA Formula 3 European Championship' has been a self-contained series that replaced the Formula 3 Euro Series. The FIA Formula 3 European Championship enters its fourth season in 2016 and has already established itself as the most significant Formula 3 series worldwide. [1]

### The Euroformula Open

Branching out from the F3 European Championship, the Euroformula Open is one of the most famous international Formula 3 championships held in Europe.

The series was established in 2001 with the name of 'Campeonato de España de Formula 3' to fill the gap after the disappearance of the rest of single-seater championships in Spain such as the Formula Renault of the Formula SuperToyota. The aim was to create a stepping stone into the bigger formula series, like the Formula V8 3.5

Since 2004 GT Sport (official organisers of the International GT Open) took control of the championship, giving the winner of the championship the chance of an official test in a Formula 1 car.

The series was renamed as 'European F3 Open' in 2009 and labelled as an official series by the FIA (Federation Internationale de l'Automobile), meaning the winner could earn the chance to have an official test day with the Scuderia Ferrari F1 Team (since the disappearance of the Toyota F1 Team in 2009) and the Super license required to drive a F1 car.

In 2014 the FIA decided to use the term 'Formula 3' only in the F3 European championship, so the series adopted its current name 'Euroformula Open'.

Today the Euroformula Open offers the best platform to enter Formula 3 racing. Being famous for its strict budget control rules that guarantee a viable option for drivers to keep competing without the backing of big sponsors, thus helping them to being promoted into bigger categories while offering premium TV covering. Additionally, budget control ensures close racing where the set-up of the car and driver skills are the main factors to achieve victory running in top-rated tracks around Europe. [2]

The championship is formed around a single-chassis, single-engine provider, the latter being different to the vast majority of other Formula 3 series and ensuring equity for all teams. There exists a B-Division inside the series (labelled as Copa de España) where teams can run the old specification chassis in order to provide an opportunity for drivers with low budget. Both categories run together on the track. [2]

The weekend format of the race is as follows [3]:

- Friday
  - Free practice sessions
- Saturday
  - Qualifying 1: 30 minutes
  - Race 1: 95 Km or 35 minutes maximum
- Sunday
  - Qualifying 2: 30 minutes
  - Race 2: 95 Km or 35 minutes maximum

### 3.1.2. The Manufacturer

Dallara Automobili is the manufacturer in charge of the design, assembly, development and production of the vehicle. Founded in Varano de' Melegari in 1972 behind the founder and current president of the company Giampaolo Dallara's house, this Italian car manufacturer is arguably the world's most successful producer of racing vehicles. [4]

Since its establishment as an independent company after the founder's work at Ferrari, Maserati, Lamborghini and De Tomaso the Dallara Formula 3 car skyrocketed the brand's success and image first in Italy then around the world.

Debuting its first Formula 3 car in 1978 and winning the Italian championship in 1980, Dallara broke out a fight between its French and English rivals for the international market. The appearance of the Dallara F-393, featuring a single-fronted shock absorber and an excellent aerodynamic balance ensured that a 90% of the Formula 3 cars in the following edition of the championship were Dallara made. [5]

Having a rule that requires a 4-year regulatory stability to contain the annual operating costs for the teams only benefited the brand, peaking the car production to more than 80 vehicles for the season and even reaching 150 in the following 3 years. [5]

The Formula 3 success allowed the company to expand into other categories and being known for their accomplishment of high standards of quality, performance, safety and customer support.

Nowadays every weekend roughly 300 Dallara cars compete in different tracks around the world. The brand has become the sole supplier of all Formula 3 championships and the sole supplier of cars for the Indycar in the United States, Indy Lights, GP2, GP3, Formula 3.5 V8, Super Formula and Renault Sport Trophy.

Additionally, in recent years, the engineering activities of the brand have expanded both for race cars and high-performance road cars. Dallara provides consultancy services for famous automotive brands such as Alfa Romeo, Audi, Ferrari, KTM, Bugatti, Lamborghini and Maserati.

## 3.2. The Car

The car model specific to study in this project is the Dallara F-317. The racing car is the latest development for the 2017 season which is based upon the platform initialized with the F-393 model mentioned above. The following figures show a sketch of the car in different views. [6]

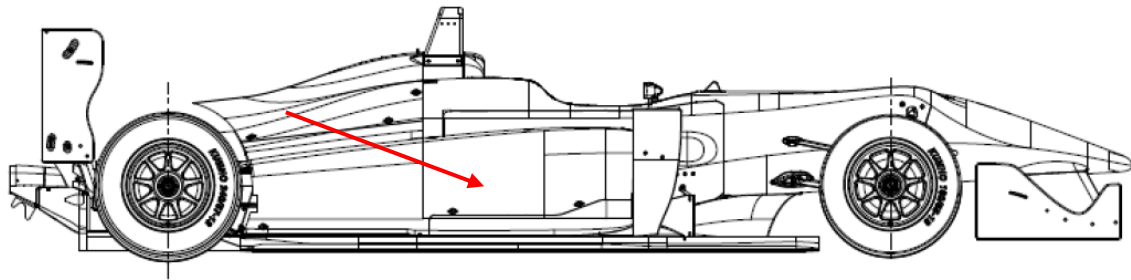


Figure 3.2: Lateral view of the Dallara F-317 (aero package shown is the one of the F-312)

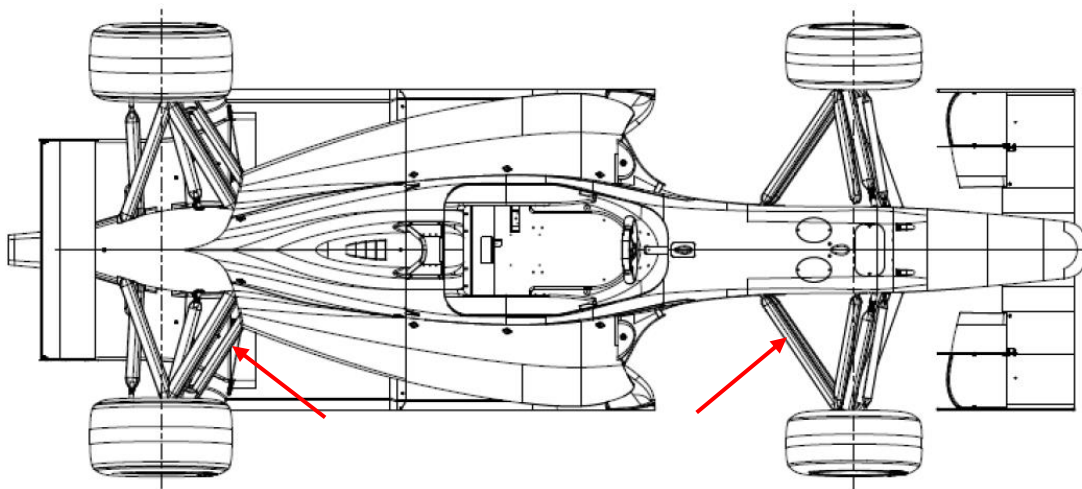
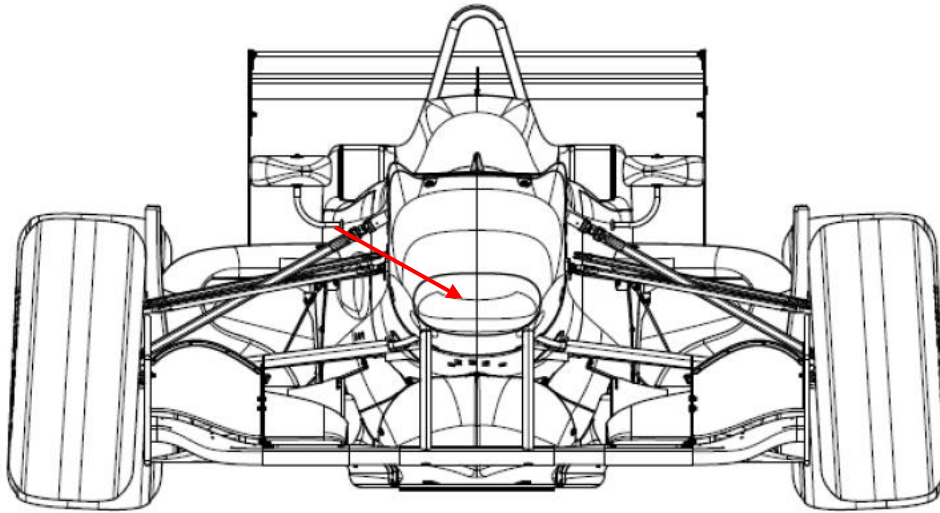


Figure 3.3: Top view of the Dallara F-317 (aero package shown is the one of the F-312)



*Figure 3.4: Front view of the Dallara F-317 (aero package shown is the one of the F-312)*

A general overview of different parts of the car is given, as well as a general table with the technical specifications of the vehicle.

The F-317 builds upon the chassis of its proven predecessor, the F-312. Thus, it received significant updates to enhance performance and safety as well as expand their homologation until the end of 2019.

In general terms, the nose has been lowered and the impact structures have been updated in accordance with the new FIA standards (Figure 3.4). Additional Zylon panels protect the survival cell from side impacts and wheel tethers sustain a 50% more of additional force (Figure 3.2). At the same time, a new aerodynamic package to boost the grip of the cars for the current season featuring new three-dimensional front wing endplates, an updated rear wing profile and a larger diffuser has ensured the 2017 season is the fastest ever in the history of the Formula 3 championship. [6]

A stepped under-floor, a crash box on both ends of the single-seater and two roll structures ensure the vehicle's safety. All the suspension and anti-roll bar elements of the car remain the same of the F-312 version (Figure 3.3). The gear shifts are executed with a longitudinally mounted sequential six-speed gearbox. Shifting events are electro-pneumatically controlled and activated by paddles on the steering wheel.

2014 marked the beginning of a new era for Formula 3, four-cylinder engines with a maximum cubic capacity of 2000 cc and direct fuel injection are used again for the current season. Series production is not stipulated. Engines must be normally aspirated. The horsepower output is limited by a 28mm air restrictor. The safety tanks made of rubber solely are fuelled with 102 octane fuel. [6]

Table 3.1 contains the general list of technical specifications of the vehicle in its default factory status

TECH SPECS DALLARA F-317		
ENGINE	TYPE	4 cylinder in-line, 4 valves per cylinder, 2 overhead camshafts
	CAPACITY	2000 cc maximum
	BORE	87.0 mm ( + 0 mm / - 1 mm)
	COMPRESSION	15:1 maximum
	HORSEPOWER OUTPUT	240 hp ( 177 kW)
	ENGINE ELECTRONICS	Bosch MS 5.8
	FUEL MANAGEMENT	Direct Injection
	WASTE-GAS CLEANING	Three-way catalyst
	LUBRICATION	Dry Sump
	DATA RECORDING	Bosch WINDarab
ENGINE MANUFACTURERS		Europe: Mercedes, Volkswagen, Oreca, NBE / Asia: Tomei, TOM's, Mercedes, Mugen, Toda.
CHASSIS	TYPE	Dallara F317
	MANUFACTURER	Dallara Automobili
	TYPE	Carbon fibre / Kevlar sandwich-structured monocoque, adjustable wings both at the front and at the rear.
	MINIMUM WEIGHT	580 kg including driver.
	WHEELBASE	2800 mm
	TRACK WIDTH	Front: 1595 mm / Rear: 1540 mm
	HEIGHT	945 mm plus ride height.
	WIDTH	1845 mm including tires.
	LENGTH	4351 mm
	SUSPENSION	Front: Push-rod actuated with twin-damper system and torsion-bar springs. / Rear: Push-rod actuated with twin damper system and coil springs.
CHASSIS	BRAKES	Disc brake with internally ventilated brake disc by Brembo.
	GEARBOX	6-speed, sequential by Hewland.

Table 3.1: Technical specifications of the Dallara F-317



Table 3.2 builds upon the above one and shows the general characteristics of the Dallara F-317 including the suppliers of the different components needed to run in compliance with the Euroformula Open rules. [7]

GENERAL CHARACTERISTICS & SUPPLIERS	
Wheelbase	2800 mm
Front Track	1595 mm
Rear Track	1540 mm
Overall Length	4351 mm
Overall Width	1845 mm (including tires)
Overall Height	945 mm (plus ride height)
Front Suspension	Pushrod twin damper system with torsion bar springs
Rear Suspension	Pushrod twin damper system with coil springs
Chassis	Carbon sandwich with aluminum/Nomex honeycomb
Bodywork	Glass fibre composite with Nomex honeycomb
Gearbox	Hewland, sequential six forward gears + reverse
G-Box internals	Hewland gears and power-flow differential
Springs	Dallara torsion abr springs and 3"/36mm ID coils
Dampers	Koni 2812 bump and rebound adjustable
Fuel Cell	Premier FT3
Extinguisher	OMP electrically operated
Steering Wheel	OMP
Quick Release	SPA-Design
Wheels	OZ 9" Front & 10,5" Rear
Brakes	Brembo
Battery	DEKA ref.DEAKETX 30L or 16
Seat Belts	OMP
Engine	3SGE Toyota, specific to Euroformula Open
Paddle Shift	Pneumatic - Piedrafita

Table 3.2: General characteristics and suppliers of the  
Dallara F-317

The Euroformula Open championship rules demand the use of a single-make and model of tyre, corresponding to the Michelin S412 slick tyre. [7]

These tyres feature a rubber treated specially for sprint race formats in a medium hardness compound. [9]

Table 3.3 table shows the suggested default initial set-up for the Dallara F-317 in its Euroformula Open compliant status. This specific configuration is the one used as the **starting point** for the studies done later.

SUGGESTED SET-UP FOR MICHELIN TYRE		
FRONT	Ride Height	14,5 mm
	Spring	21,5 Nm/deg
	Spring Pre-Load	0 to 0,5 turn
	Pushrod Length	Pushrod adjuster
	Roll Center Setting	Std.
	ARB	65 Kg/mm
	Camber	4,5°
	Caster	16°
	Toe	3° (toe out)
REAR	Ride Height	36 mm
	Spring	950 Lb/In
	Spring Pre-Load	No
	Pushrod Length	Pushrod adjuster
	Roll Center Setting	R-1
	ARB	90 Kg/mm
	Camber	3,5°
	Toe	4° (toe in)
	Differential	80/80 4F

Table 3.3: Suggested setup for the Michelin S412  
tyre for the Dallara F-317

### 3.3. Suspension system

In general terms, a suspension system connects the unsprung mass of the vehicle (the tyres) with the sprung mass (the chassis). This connection does not only dictate the path of relative motion, but also controls the forces that are transmitted between them. It serves a dual purpose, contributing to the car's handling and braking.

Any particular geometry of a suspension system must be designed to meet the needs of the particular vehicle for which it is to be applied. There is no single best geometry. [10]

In general terms a suspension system has the following objectives:

- Support of the weight of the car
- Provide a smooth ride
- Allowance of rapid cornering without extreme body roll
- Keep tires in firm contact with the road
- Allowance to the front wheels to turn side-to-side for a correct steering manoeuvre.
- Together with the steering system, keep the wheels with the correct alignment
- Isolation of the passengers from vibrations and shock

In motorsports these objectives are still true, except for the passenger's isolation from vibrations and the providence of a smooth ride, as it is more important to maximize the contact of the tyre with the tarmac and the cornering speed to achieve faster lap times.

Suspension systems can be broadly classified into three subgroups: dependent, independent and semi-independent. These terms refer to the ability of opposite wheels moving independently of each other.

A dependent suspension generally has a beam that holds wheels parallel to each other and perpendicular to the axle. When the inclination angle of one of the wheels changes, the inclination angle of the opposite wheel changes as well. In the same manner, when one wheel hits a bump in the road its upward movement causes a tilt on the other wheel.

An independent suspension allows wheels to rise and fall on their own without affecting the opposite wheel.

A third type of suspension systems is a semi-dependent suspension. In this case, the motion of one wheel does affect the position of the other but they are not rigidly attached to each other.

Almost all single-seater racing vehicles use independent suspension systems and the Dallara F-317 is not an exception, so this is the specific system in which the project is focused from this point onward.

### 3.3.1. Independent suspension systems: the double wishbone

For an independent suspension, be it front or rear, the assemblage of control arms is intended to control the wheel motion relative to the car body in a single prescribed path. In engineering terms, it can be said that the wheel has a fixed path of motion relative to the car body. It is not possible to move fore and aft or laterally relative to this path (Figure 3.5). The suspension linkages are expected to position the wheel very accurately in all directions while allowing it to move up and down against the spring and shock. [11]

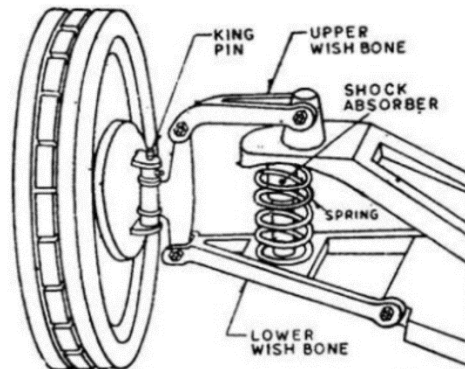


Figure 3.5: A typical double wishbone independent suspension attached to the wheel system

A single body has six degrees of freedom in a three-dimensional space (Figure 3.6). Any independent suspension system provides five movement restraints, allowing only the one path of motion of the wheel relative to the car body.

Therefore, the goal of the study of independent suspension geometries is to determine how to restrain the motion path of the wheel to an optimum motion path.

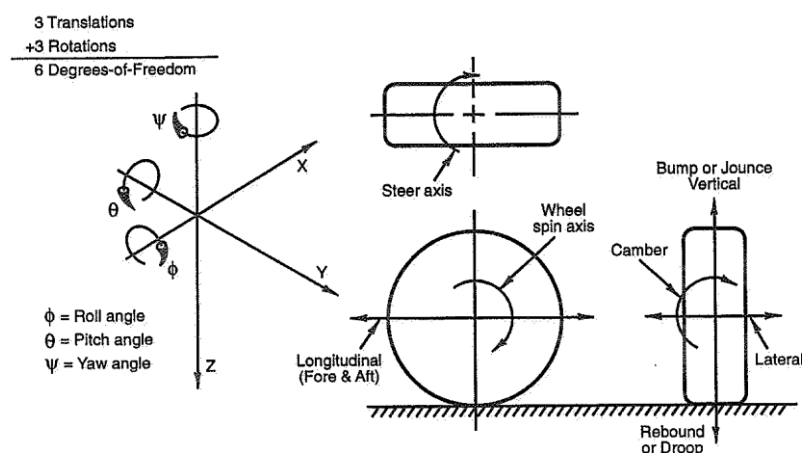


Figure 3.6: The six degrees of freedom a wheel can have in space and the different rotations and movements it can have.

The standard racing double wishbone suspension has two A-arms plus a tie rod. Thus, two links for each A-arm and one link for the tie rod adds up to five (Figure 3.7).

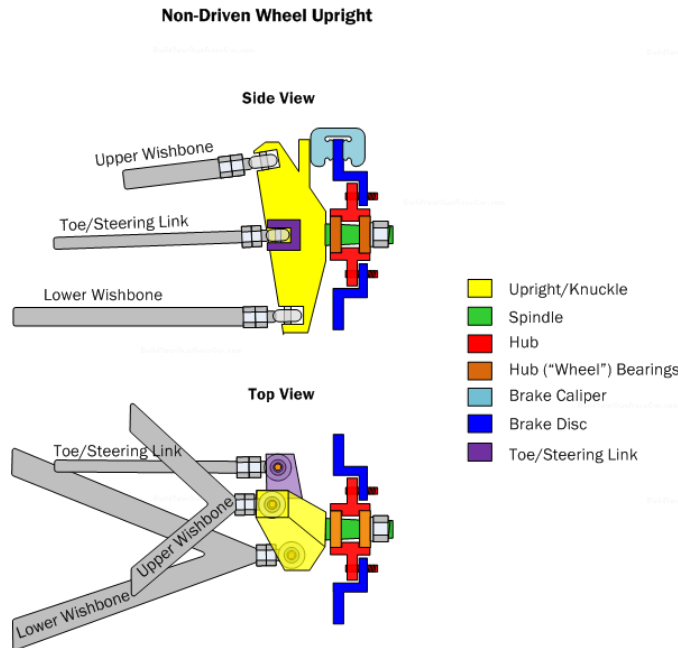


Figure 3.7: Elements of a general double wishbone racing car suspension geometry

The double wishbone suspension system is widely regarded as one of the best the best solutions among the available alternatives for a single seater due to its lightweight, easy-to-operate, easy-set-up capabilities. In addition, a double wishbone suspension system offers simple geometry, avoiding bulky pieces and allowing the air to flow easier from the front to the back of the chassis, improving the aerodynamic balance of the car; a point of big relevance in single-seater racing. Two different geometries for a double wishbone are the most typically found in single-seaters: push rod and pull rod arrangement.

#### Push Rod vs Pull Rod arrangement

There exist different opinions over which of the two types of these double wishbone, independent suspension systems are better, but in reality, both have pros and cons and each type is more suitable than the other depending on the specific needs of the application. [12]

The only difference in the geometric arrangement of the elements between these two systems is that the pull rod suspension arm works under tensile stress and the push rod suspension arm works under compression stress (Figure 3.8):

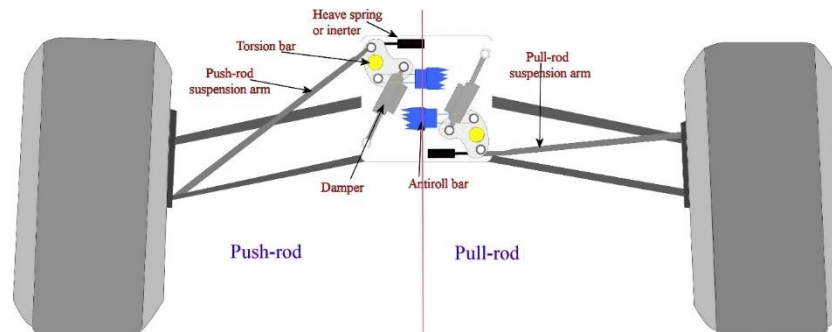


Figure 3.8: Push rod and pull rod arrangement comparison.

As Figure 3.8 shows, a pull-rod suspension system allows to position the spring, damper, rocker, anti-roll bar package lower, thus lowering the overall centre of gravity of the car, which is an advantage compared to the push-rod configuration. However, positioning the above-mentioned package in the upper part of the car makes accessing all the elements easier for a faster set-up or replacement.

The push-rod system has a slight advantage over the pull-rod system aerodynamically-wise. Because of its geometry the air can be channelled easier compared to a pull-rod configuration.

Traction efforts are better supported by the arm in a pull-rod configuration, allowing for a lighter construction to be used. However, a pull-rod system generally puts a higher load on the upper control arm. [12]

The Dallara F-317 uses push-rod geometries in both the front and rear axles of the car due to its easy operability and faster set-up capabilities. Figure 3.9 shows the car's suspension system.



Figure 3.9: Dallara F-317 image where the push rod arrangement can be seen in both axles.

### 3.4. Dallara F-317's suspension system

The Dallara F-317 uses a push-rod suspension system in both axles as described in the previous section. Figures 3.10 and 3.11 show the corresponding right wheel of both axles respectively. It is important to note that the list of numbers marked in both of the figures are intended only to situate the coordinates of all the elements of the suspension system in space. The coordinates are listed in Table 3.4.

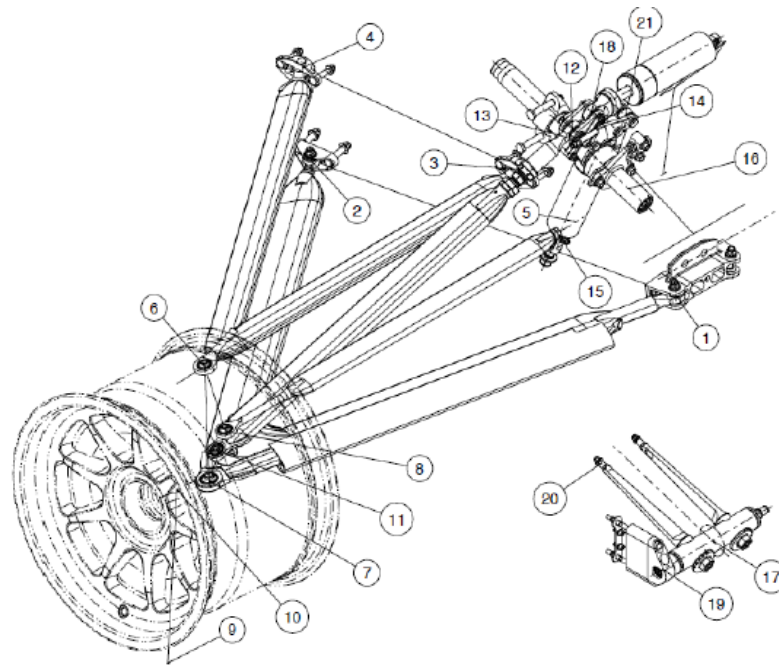


Figure 3.10: Dallara F-317's front axle suspension system

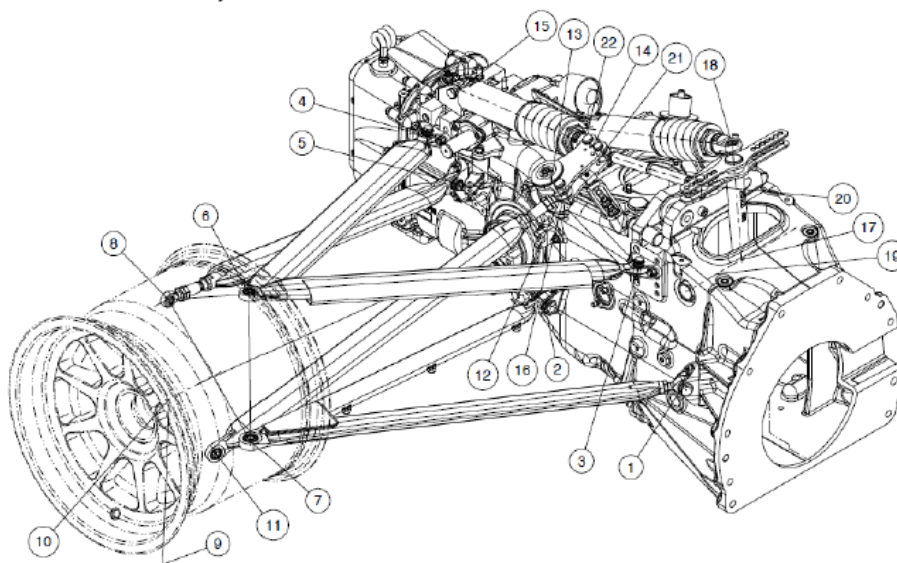


Figure 3.11: Dallara F-317's rear axle suspension system

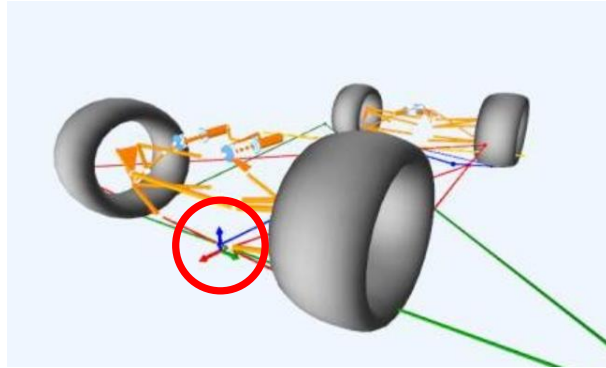


Table 3.4 describes the geometric coordinates in space of the different elements of the suspension system of the car for both axles. As stated above, the numbers are associated with the ones appearing in the elements of both axle's suspension systems in figures 3.10 and 3.11.

FRONT / DELANTERA	X [mm]	Y [mm]	Z [mm]	REAR / TRASERA	X [mm]	Y [mm]	Z [mm]
P1	190,5	-50	282	P1	415	-150	167
P2	-380	-153	263	P2	121,9	-134	155,5
P3	17	-162,4	452	P3	309	-135	308,1
P4	-380	-168,8	418,5	P4	-156	-90	279
P5	127	-170	427,5	P5	-92,5	-101	237
P6	-65,9	-663,7	347,3	P6	25	-623,5	364,5
P7	-23,9	-697,2	201,8	P7	85	-680	170
P8	31,5	-723,0	322,3	P8	-122	-637	269
P9	0	-797,5	-26,5	P9	0	-770	-39,5
P10	0	-781,1	242,0	P10	0	-757,1	238,7
P11	-32,2	-678,6	239,4	P11	12	-678	113
P12	91,5	-141,6	542,0	P12	164,5	-138,2	368,7
P13	82,4	-125	512,9	P13	129,1	-128,1	364,7
P14	79,2	-58,5	502,7	P14	142,4	-59,7	395,1
P15	21,0	-96,5	317,3	P15	-143,4	-37,7	339,2
P16	177,8	-125	482,9	P16	137,5	-113,5	328,4
P17 (int)	115,5	0	364,2	P17	380	0	290
P17 (ext)	173,5	0	346,2	P18	361,7	0	477,1
P17 (ext +40)	213,5	0	346,2	P19	380	-50	290
P18	96,3	-111,5	557,3	P20	365	-66	443,108
P19 (int)	115,5	-93,2	364,2	P21	168,3	-65,2	398,9
P19 (ext)	173,5	-93,2	346,2	P22	75,0	0	362,7
P19 (ext +40)	213,5	-93,2	346,2				
P20	45,6	-35	400,6				
P21	79,6	-32	504,1				

Table 3.4: List of coordinates for the elements of the front and rear suspension system represented in figures 3.10 and 3.11

It is very important to note for the future use of these coordinates that the manufacturer establishes the origin of the coordinate system independently for both axles. In both cases it is situated in the intersection of the longitudinal symmetry plane of the car and the line that follows the axle of the car from the centered contact point with the ground of the right wheel to the centered contact point with the ground of the left wheel. Figure 3.12 shows the point where the front axle's origin of coordinates is located.



*Figure 3.12: Representation of the point defined by the manufacturer as the front axle's origin of coordinates*

Using the data shown in table 3.4 and the origin of coordinates, the geometrical configuration of the suspension system can now be introduced in Optimum Kinematics to construct the starting point of the car for further analysis.



### 3.5. Modelation and simulation of the fore and aft suspension system

Optimum Kinematics is a vehicle kinematics software that provides a user-friendly layout, interactive 3D visualizations, a fast and reliable solver, direct application to car design and setup and accuracy in the results with more that 400 data channels that can be plotted against each other. [13] The default structure of the program is used to model and simulate the configuration of the fore and aft suspension system. Therefore, the process is divided in the categories shown in Figure 3.13

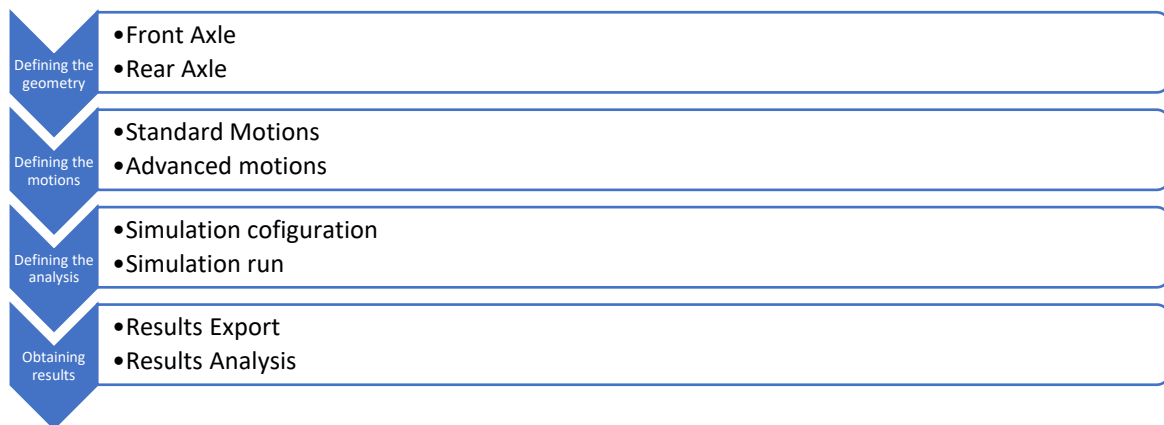


Figure 3.13: Analysis process of the study

### Reference system in Optimum Kinematics

It is important to define the reference system the software uses before starting to create the geometry of the configuration of the suspension system and simulating it. The reference system for the direction and rotation axes is defined in Table 3.5, while Figure 3.14 is a graphical representation that supports the definitions stated on the table.

Axis	Defintion	Name	Color
Longitudinal	Positive when it points to the front of the car	X	Red
Lateral	Positive when it points to the left of the car	Y	Green
Vertical	Positive when it points vertically up	Z	Blue
Roll	Counter Clockwise positive	Mx	Red
Pitch	Counter Clockwise positive	My	Green
Yaw	Counter Clockwise positive	Mz	Blue

Table 3.5: Types of axis in Optimum Kinematics and its definition

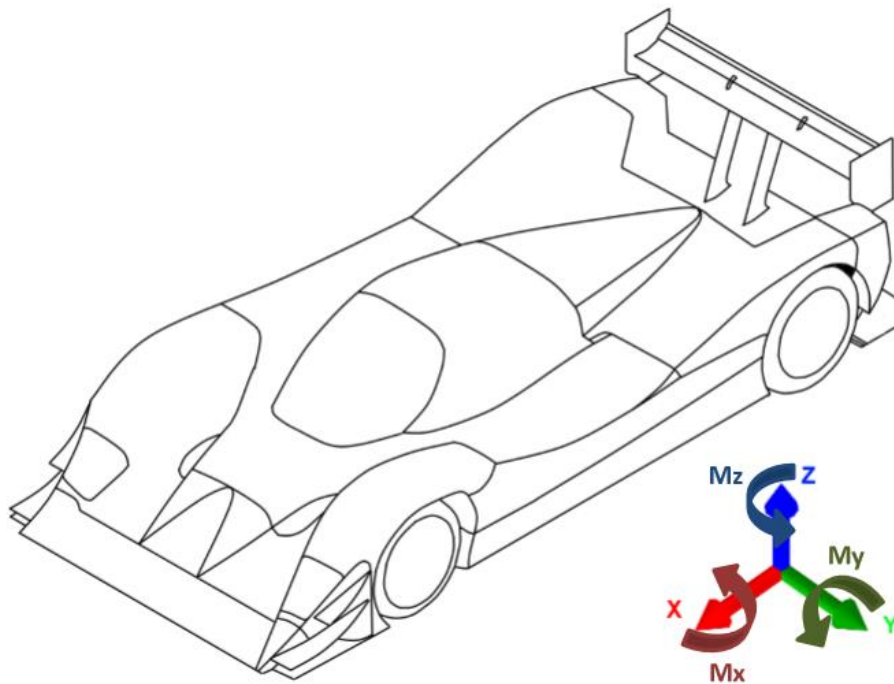


Figure 3.14: Graphical representation of Optimum Kinematics' axes

This reference system is to be used throughout the complete process of the analysis for the starting configuration as well as for the customized configuration.

### 3.5.1. Defining the geometry

This process defines the coordinates in space of the significant elements of the suspension system using the data gathered from section 3.4. Therefore, both axles can be modelled. The different categories and coordinates entered to do model them are explained below. Figure 3.16 offers graphical information of the input data box in Optimum Kinematics and how to introduce the coordinates.

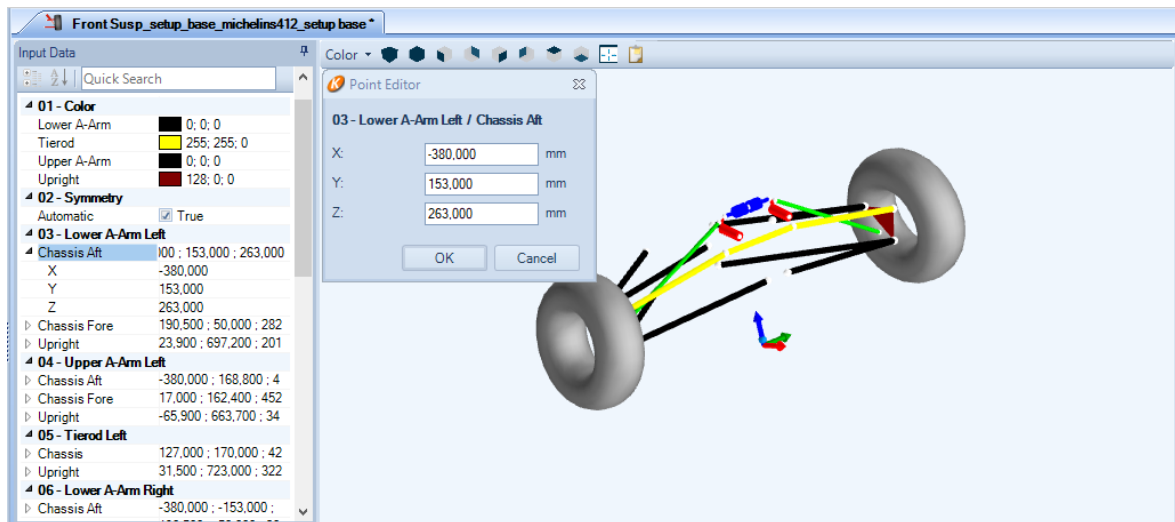


Figure 3.15: Introduction of coordinates in Optimum Kinematics

#### FRONT AXLE:

The front axle's geometry of the car is defined in optimum by the Double A-Arm, the rack and pinion, the wheels and the torsion bar.

#### Double A-Arm:

	Lower A-Arm Left			Upper A-Arm Left			Tierod Left	
	Chassis Aft	Chassis Fore	Upright	Chassis Aft	Chassis Fore	Upright	Chassis	Upright
X	-380,0 mm	-190,5 mm	23,9 mm	-380,0 mm	17,0 mm	-65,9 mm	127,0 mm	31,5 mm
Y	153,0 mm	50,0 mm	697,2 mm	168,8 mm	162,4 mm	663,7 mm	170,0 mm	723,0 mm
Z	263,0 mm	282,0 mm	210,8 mm	418,5 mm	452,0 mm	347,3 mm	427,5 mm	322,3 mm

Table 3.6: Double A-Arm coordinates introduced in Optimum Kinematics

The symmetry option is marked as shown below in Figure 3.16, obtaining all the coordinates for the right side of the car.

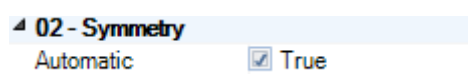


Figure 3.16: Symmetry option in Optimum Kinematics

### Rack and Pinion:

In this category, the steering ratio of the car is defined. Using the technical sheet provided by the manufacturer, the data from Figure 3.17 is known:

Pinion primitive diameter	15.60 mm
Static steering ratio	12.5° steering wheel/1°wheel
Ackermann [%]	28

Figure 3.17: Steering Ratio of the Dallara F-317 as defined in the User's Guide

It is possible to use it to calculate the steering ratio as Optimum Kinematics demands it by multiplying the pinion's primitive diameter by the number pi to get the longitudinal steering rack displacement in a complete revolution of the pinion, which equals a full revolution of the steering wheel.

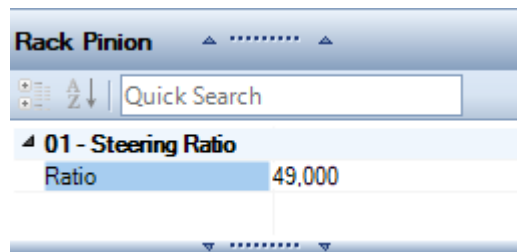


Figure 3.18: Steering Ratio in Optimum Kinematics' input data box

### Wheels:

Using the data extracted from the Michelin Motorsport website and following the recommended setup for the Michelin tyre (section 3.2) the following fields are filled:

- Left Tyre:
  - Half Track: 797,5 mm
  - Offset Lateral: 0 mm
  - Offset Longitudinal: 0 mm
  - Offset Vertical: 0 mm
  - Rim Diameter: 228,6 mm
  - Static Camber: -4,5°
  - Static Toe: 3,0°
  - Tire Diameter: 540,0 mm
  - Tire Width: 210,0 mm

Using the symmetry option, the data for the right tyre is completed.

### Torsion Bar:

The torsion bar system used at the front can be defined using the categories provided by Optimum Kinematics. The coordinates introduced are shown in Table 3.7

	Push Rod Left		Rocker Left		Damper Left	
	Non-Suspended mass	Rocker	Axis	Pivot	Chassis	Rocker
X	-32,2 mm	91,5 mm	177,8 mm	82,4 mm	96,3 mm	96,3 mm
Y	678,6 mm	141,6 mm	125,0 mm	125,0 mm	10,0 mm	111,5 mm
Z	239,4 mm	542,0 mm	482,9 mm	512,9 mm	557,3 mm	557,3 mm

Table 3.7: Coordinates of the Torsion bar introduced in Optimum Kinematics

The symmetry option can now be applied to obtain the values for the right side.

Lastly, the attachment place of the torsion bar system must now be defined. In the case of the vehicle of study, the torsion bar is attached to the double A-arm's upright.

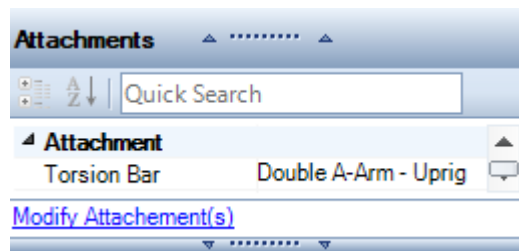


Figure 3.19: Attachment input data box for the front axle's suspension system

The front axle's geometry of the suspension system configuration is now complete. Figure 3.20 shows a representation of the geometry of the front axle of the Dallara F-317 in Optimum Kinematics.

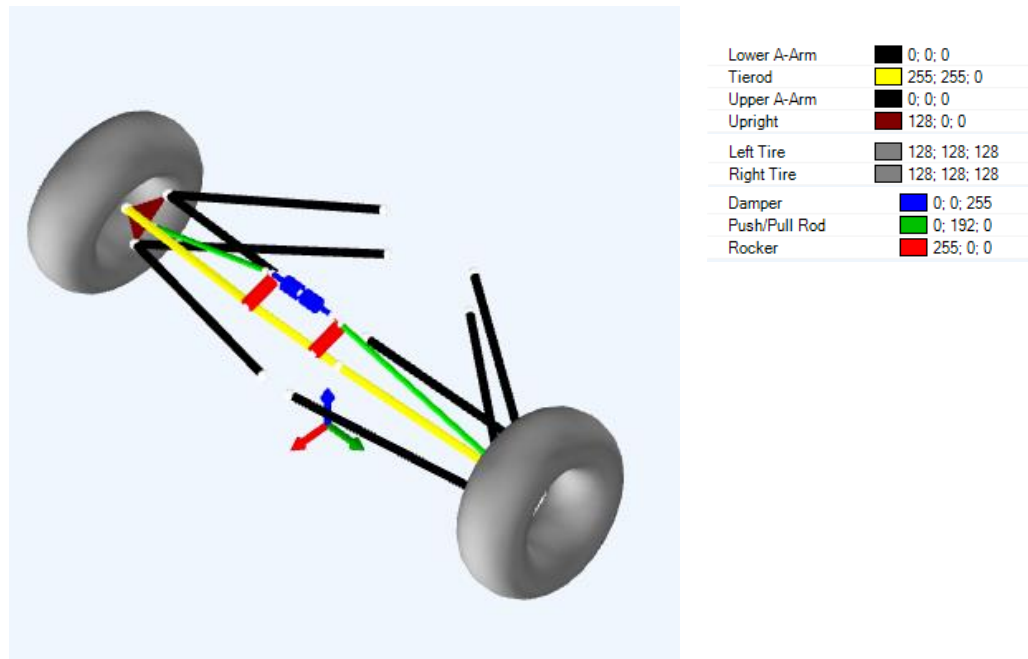


Figure 3.20: Front axle's geometry represented in Optimum Kinematics

### REAR AXLE:

The geometry of the rear axle suspension system is defined in Optimum Kinematics by the double A-arms, the wheels, and the pushrod and coils system.

#### Double A-Arm:

Using the same procedure as with the front axle, the coordinates to define the double A-arms of the rear axle suspension system are introduced into the categories of Table 3.8:

	Lower A-Arm Left			Upper A-Arm Left			Tierod Left	
	Chassis Aft	Chassis Fore	Upright	Chassis Aft	Chassis Fore	Upright	Chassis	Upright
X	121,9 mm	415,0 mm	85,0 mm	-156,0 mm	309,0 mm	25,0 mm	-92,5 mm	122,0 mm
Y	134,0 mm	150,0 mm	680,0 mm	90,0 mm	135,0 mm	623,5 mm	101,0 mm	637,0 mm
Z	155,5 mm	167,0 mm	170,0 mm	279,0 mm	308,1 mm	364,5 mm	237,0 mm	289,0 mm

Table 3.8: List of coordinates introduced in Optimum Kinematics of the different linkages of the rear axle's double A-Arm

Just like in the front axle, by using the automatic symmetry option, the right side of the rear double A-Arm is generated.

### Wheels:

Using the data extracted from the Michelin Motorsport website and following the recommended setup for the Michelin tyre (section 3.2) the following fields are filled:

- Left tyre:
  - Half Track: 770,000 mm
  - Offset Lateral: 0 mm
  - Offset Longitudinal: 0 mm
  - Offset Vertical: 0 mm
  - Rim Diameter: 266,700 mm
  - Static Camber: -3,500 mm
  - Static Toe: -4,000 mm
  - Tire Diameter: 574,000 mm
  - Tire Width: 260,000 mm

Using the automatic symmetry option, the right tyre of the rear axle is automatically defined.

### Pushrod and springs:

The pushrod-spring system of the rear axle is defined in Optimum Kinematics filling the coordinates specified in Table 3.9:

	Attachment Left		Rocker Left		Coilover Left	
	Non-Suspended mass	Rocker	Axis	Pivot	Chassis	Rocker
X	12,0 mm	164,5 mm	137,5 mm	129,1 mm	-143,4 mm	142,4 mm
Y	678,0 mm	138,2 mm	113,5 mm	128,1 mm	37,7 mm	59,7 mm
Z	113,0 mm	368,7 mm	328,4 mm	364,7 mm	339,2 mm	395,1 mm

Table 3.9: List of coordinates introduced in Optimum Kinematics of the different linkages of the rear axle pushrod-spring system.

Again, using the automatic symmetry option, the right side of the rear pushrod system is automatically generated.

Lastly, the place of attachment of the Pushrod must be defined. In the case of the Dallara F-317, it is attached to the upright.

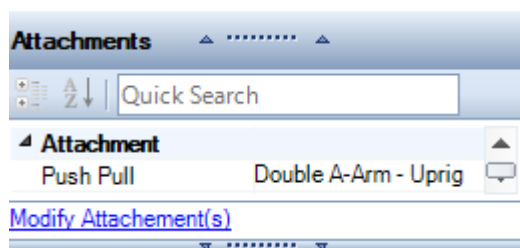


Figure 3.21: Attachment input data box for the rear axle's suspension system

After this, the rear axle of the Dallara F-317 is defined. Figure 3.22 shows the complete rear axles together with a colour legend for each of the parts of the system.

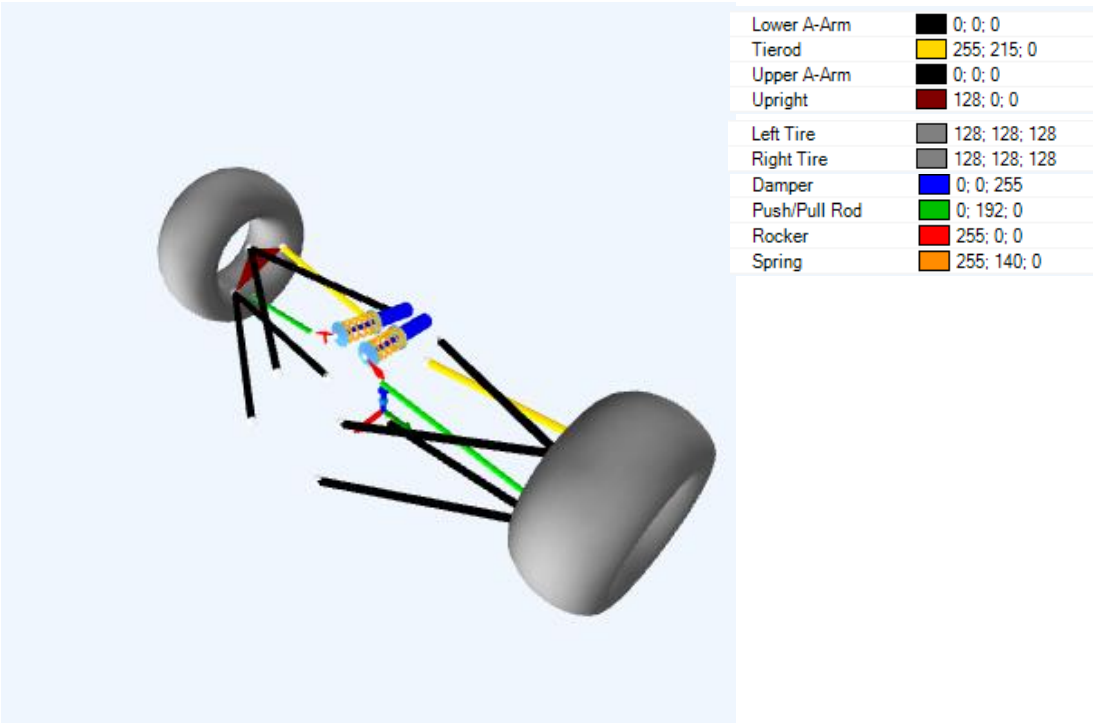


Figure 3.22: Rear axle's suspension geometry in Optimum Kinematics

The next step is to create a full vehicle, which is done selecting both of the previously defined axles separated by the wheelbase distance gathered in the technical specification table (Section 3.2, Table 3.2)

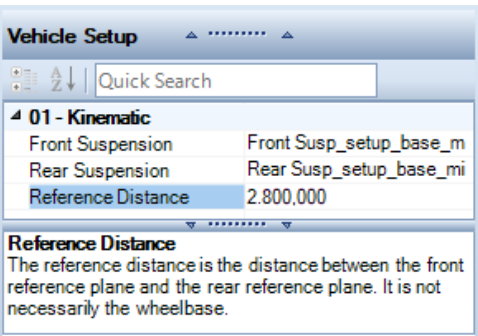
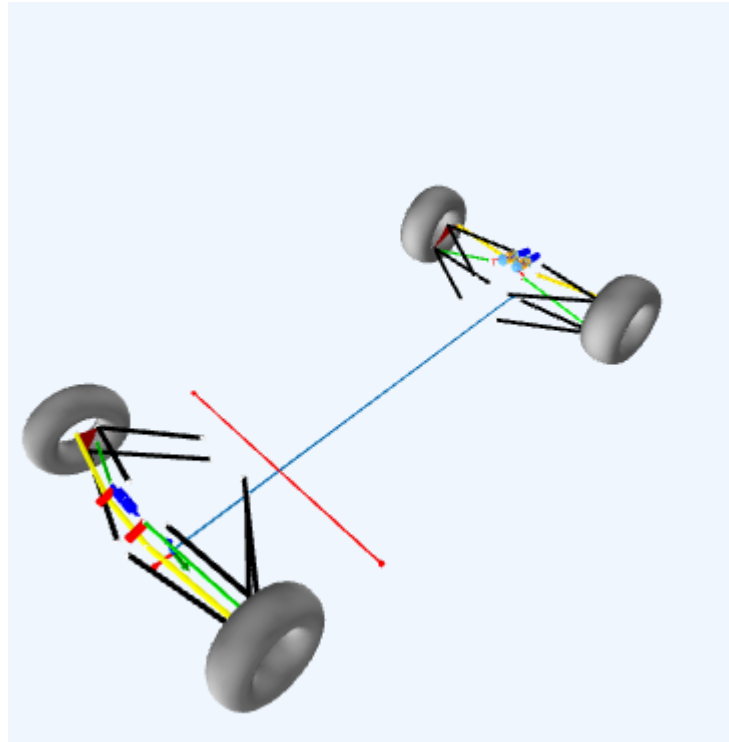


Figure 3.23: Definition of a full vehicle input data box in Optimum Kinematics



Figure 3.24 shows the representation of a full vehicle in Optimum Kinematics where the front and rear axle's suspension system configurations defined above are represented. The blue longitudinal line represents the roll center axis of the car, and the red transversal line represents the pitch center axis of the car.

The geometry of the suspension system of the Dallara F-317 is complete.



*Figure 3.24: Full Vehicle representation in Optimum Kinematics*

### 3.5.2. Defining motions

Once the geometry of the suspension system of the car is defined, the manoeuvres the car performs in the study are to be defined.

Roll:

A pure body roll of 2 degrees both clockwise and counter-clockwise is defined to simulate the chassis' behaviour during a cornering event. Figure 3.25 shows said conditions introduced in Optimum Kinematics' menu:

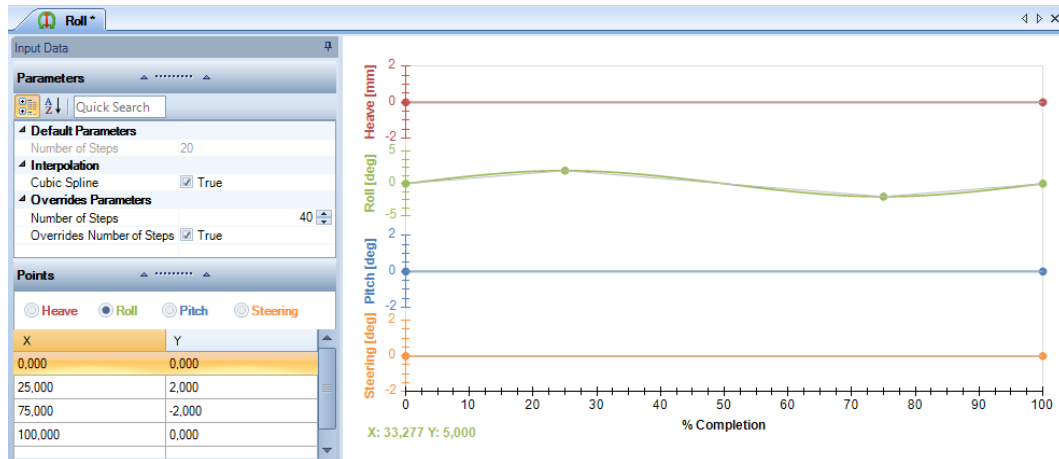


Figure 3.25: Definition of roll motion in Optimum Kinematics

Heave:

A pure heave motion of 30 mm both up and down on the vertical axis of the car is defined to study the behaviour of the vehicle during a bump event. Figure 3.26 shows said conditions introduced in Optimum Kinematics' menu:

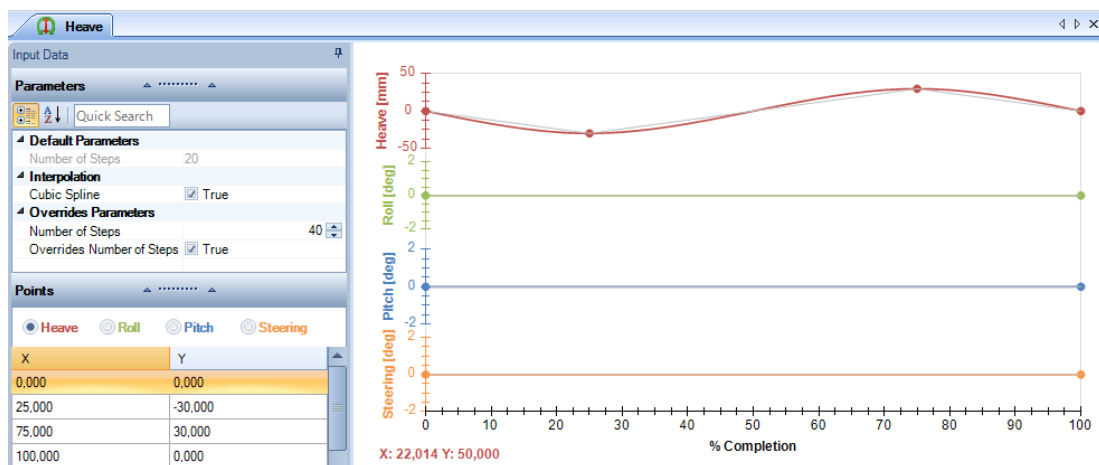


Figure 3.26: Definition of heave motion in Optimum Kinematics

### Pitch:

A pure pitch motion of 1,5 degrees both clockwise and counter-clockwise is defined to study the behaviour of the vehicle during a braking/acceleration event. Figure 3.27 shows said conditions introduced in Optimum Kinematics' menu:

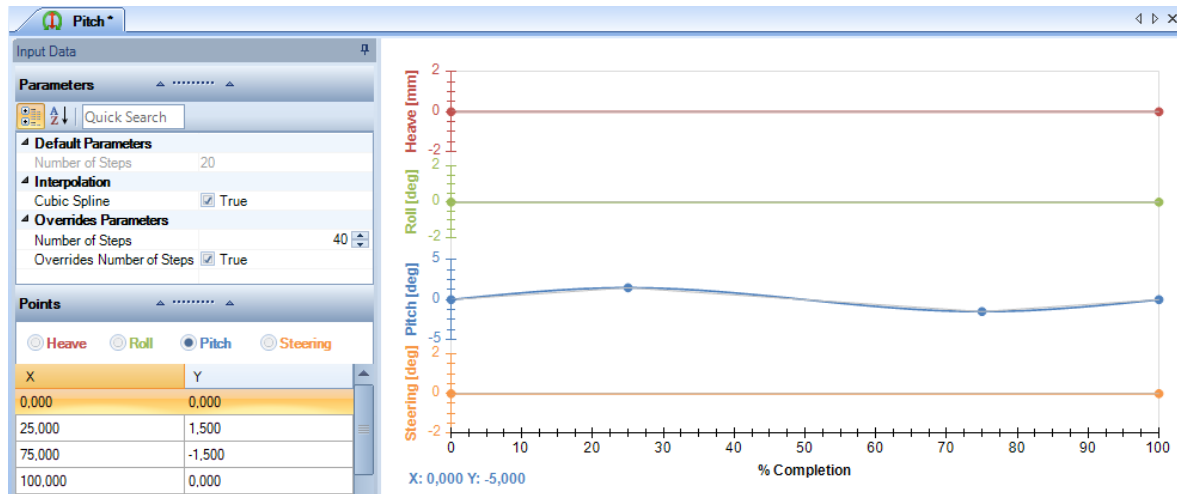


Figure 3.27: Definition of pitch motion in Optimum Kinematics

### Steering:

A basic steering motion is defined to simulate a turning event. Figure 3.28 shows the conditions introduced in Optimum Kinematics' menu:

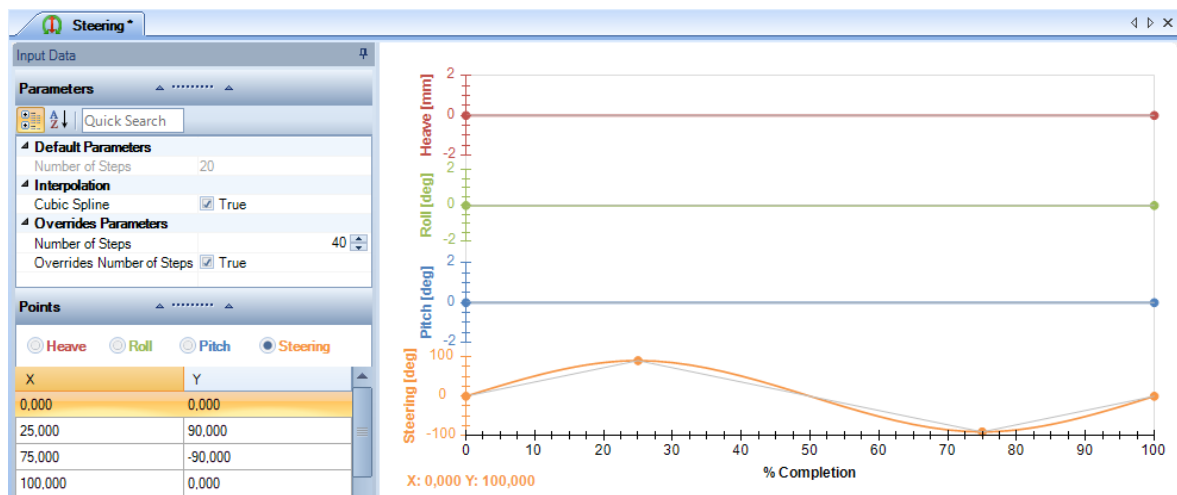


Figure 3.28: Definition of steering motion in Optimum Kinematics

### Corner Entry:

Using data provided by the mechanics of F3 Team Carlin, a corner entry move is defined combining 0,916 degrees of clockwise body roll and 0,440 degrees of clockwise body pitch. The aim is to reproduce the situation the car experiments when breaking and turning into a corner.

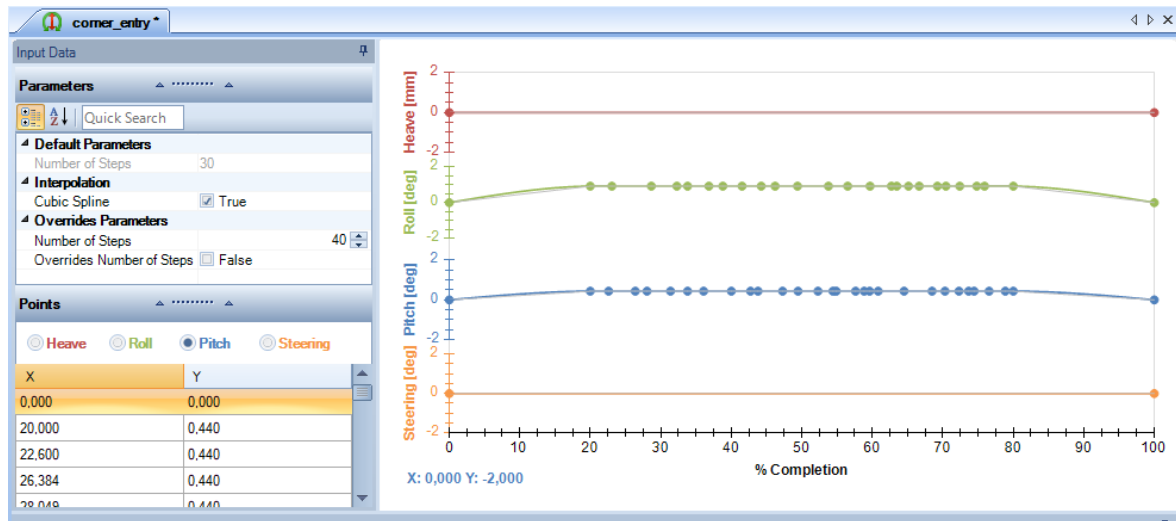


Figure 3.29: Definition of corner entry motion in Optimum Kinematics

### Corner Exit:

Using data provided by mechanic of F3 Team Carlin, a corner exit move is defined combining 0,916 degrees of clockwise body roll and -0,440 degrees of counter-clockwise body pitch. The aim is to reproduce the situation the car experiments when accelerating and exiting a corner.

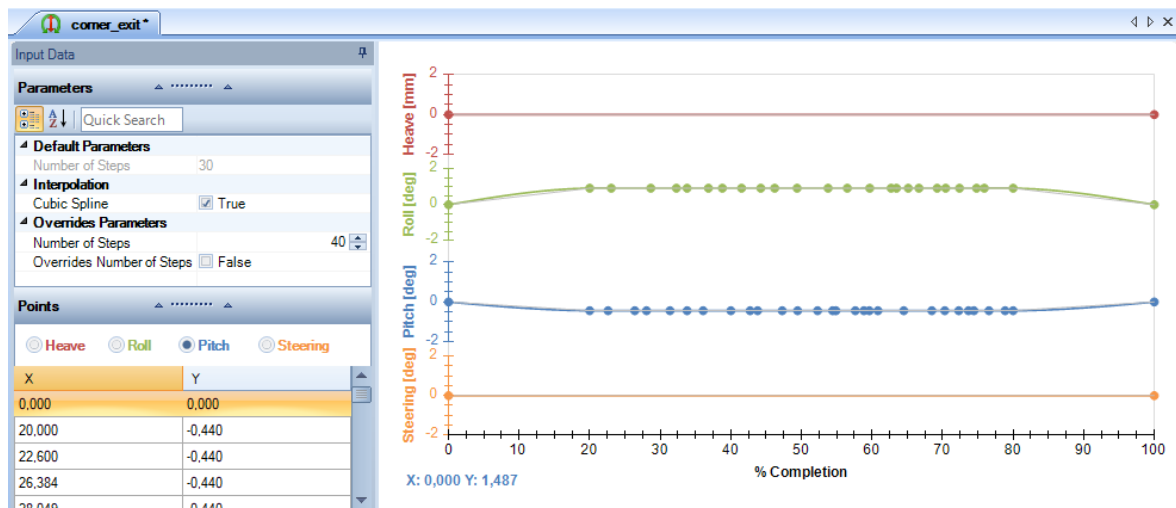


Figure 3.30: Definition of corner exit motion in Optimum Kinematics

### 3.5.3. Defining the analysis

Configuring the simulations in Optimum Kinematics is done assigning a desired, designed motion to the desired, designed vehicle.

A new simulation is created for each of the previously defined motions in section 3.5.2, assigning to all of them the baseline vehicle geometry defined in section 3.5.1. Therefore, 6 different simulations are performed.

Figure 3.31 defines the simulation configuration menu in Optimum Kinematics for the roll motion. The same process is applied to the rest of the motions defined in section 3.5.2.

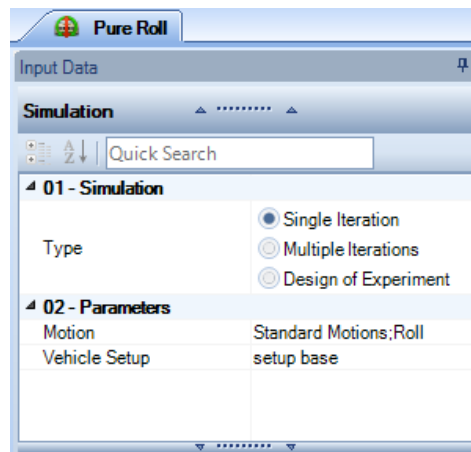


Figure 3.31: Example of the simulation configuration menu in Optimum Kinematics.

The 'Run' button shown in Figure 3.32 executes the simulation, generating an analysis file.



Figure 3.32: Layout of the main buttons of the simulation menu in Optimum Kinematics

### 3.5.4. Generating results

Once the simulation is run a new analysis file is generated under the results category.

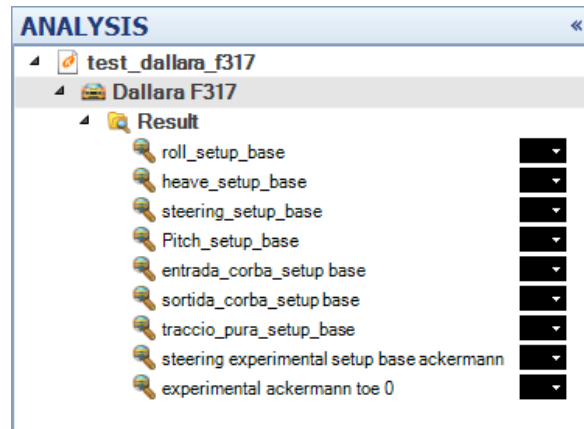


Figure 3.33: Example of the Results window in Optimum Kinematic

New reports can now be created. Each of them offering a workspace where several types of graphics and tables can be plotted using the analysis file generated after the simulation and choosing between the different data channels available (Figure 3.34)

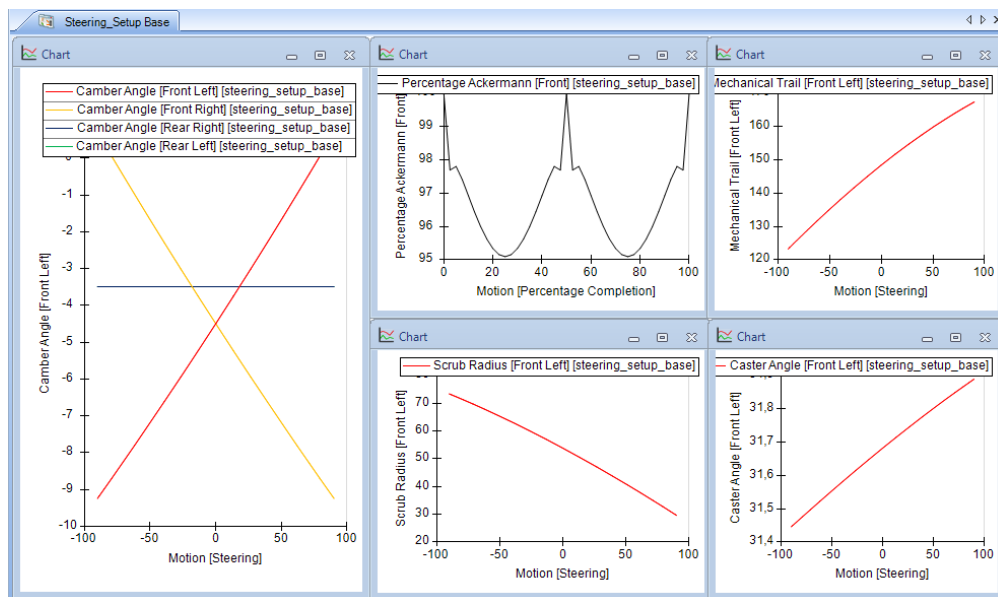


Figure 3.34: Example of a graphics tab for analysis in Optimum Kinematics

### 3.6. Base setup analysis

Now that the geometry of the suspension system of the Dallara F-317 is defined and the simulations executed using the defined motions, the analysis of the starting point configuration follows using the analysis files generated in the simulation of the 6 motions with the vehicle's baseline geometry of the suspension system.

However, previous to the analysis, a list of kinematic and geometric concepts related to the suspension system are explained in order to better clarify the results presented later and in further sections of the analysis.

Camber Angle:

Camber is defined as the inclination angle between the side plane (vertical-longitudinal plane) and the rim plane lying on the centreline of the rim. Positive camber is defined as the tops of the wheels tipping away from the vehicle.

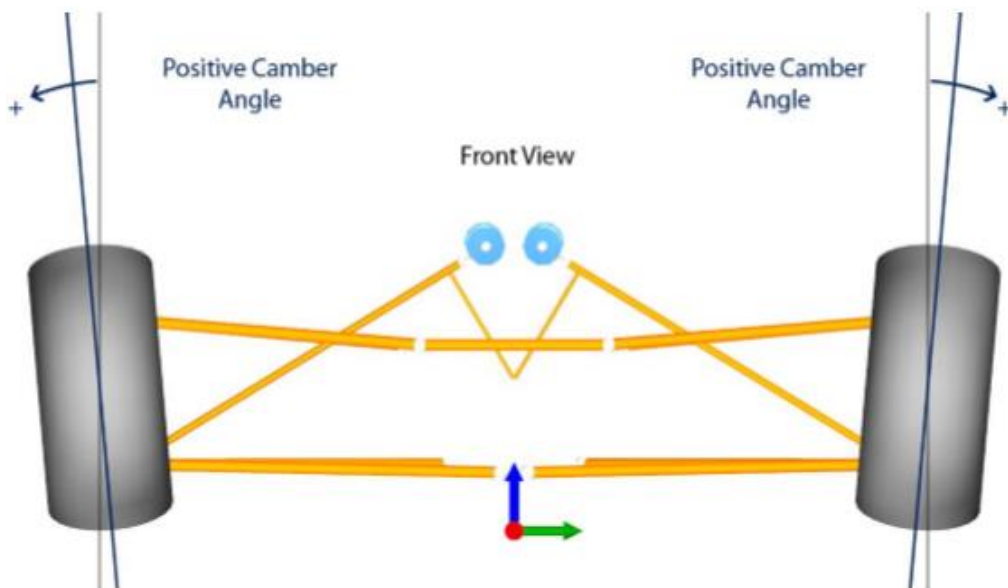


Figure 3.35: Camber angle representation in Optimum Kinematics

### Toe Angle:

Toe is defined as the angular deflection from the vehicles centreline and the centreline of the rim. Positive toe (toe out) is defined as a wheel splaying out from the direction of travel. Toe Angle carries the same sign as Toe Distance.

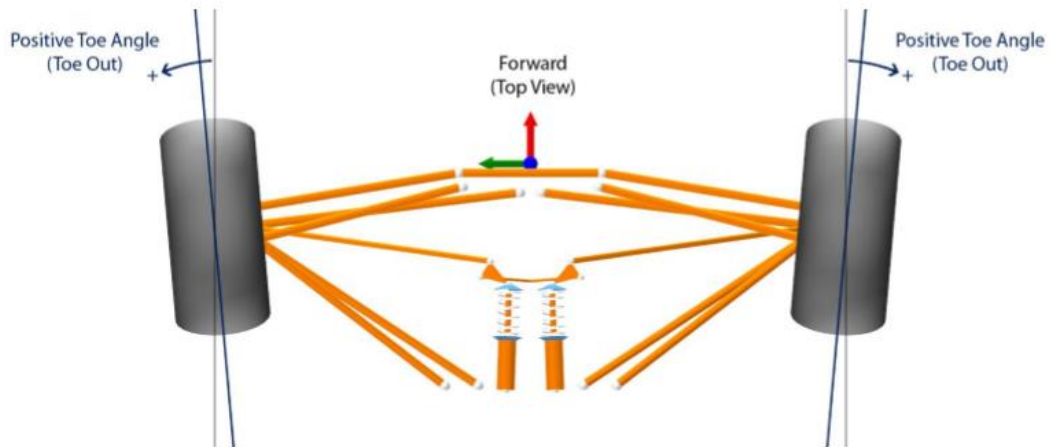


Figure 3.36: Toe angle representation in Optimum Kinematics

### Toe Distance:

Toe distance is defined as the total lateral distance between the leading and trailing edge of one side of a vehicles rim. Toe distance is taken to have the same sign as the angular toe measurement. Positive toe angle equates to positive toe distance.

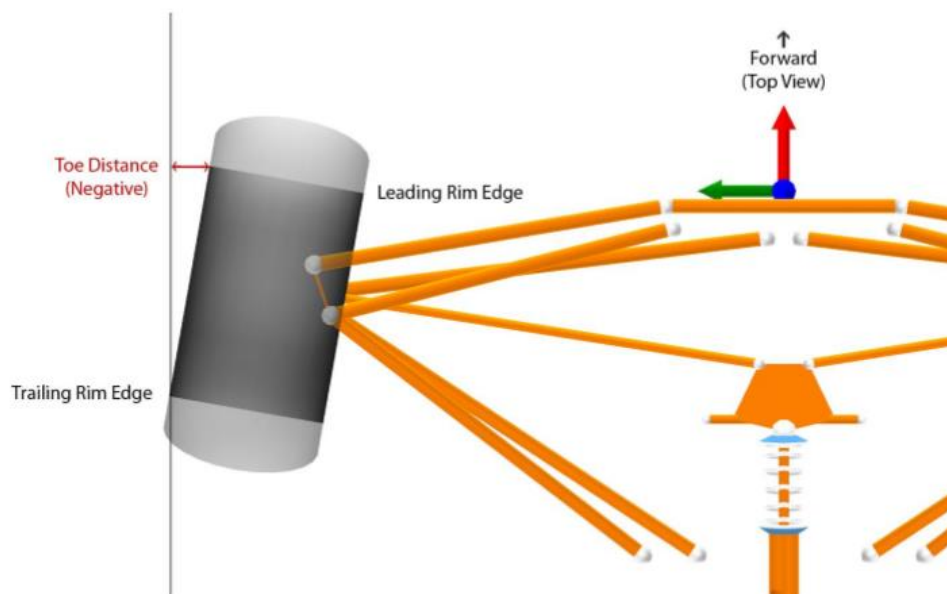


Figure 3.37: Toe distance representation in Optimum Kinematics



### Caster Angle:

Caster is defined as the angle between the steering axis and the wheel centreline extending perpendicular from the contact patch, viewed perpendicular to the side view (vertical longitudinal plane).

Positive caster is defined as the steering axis tilting back from the wheel centreline in side view (perpendicular to the longitudinal-vertical axis). (Figure 3.38)

### Mechanical Trail:

Mechanical Trail is defined as the distance between the intersection of the steering axis and the ground measured to the center of the contact patch, viewed perpendicular to the vertical longitudinal plane.

Positive mechanical trail is defined as the steering axis intersecting the ground plane before the contact patch.

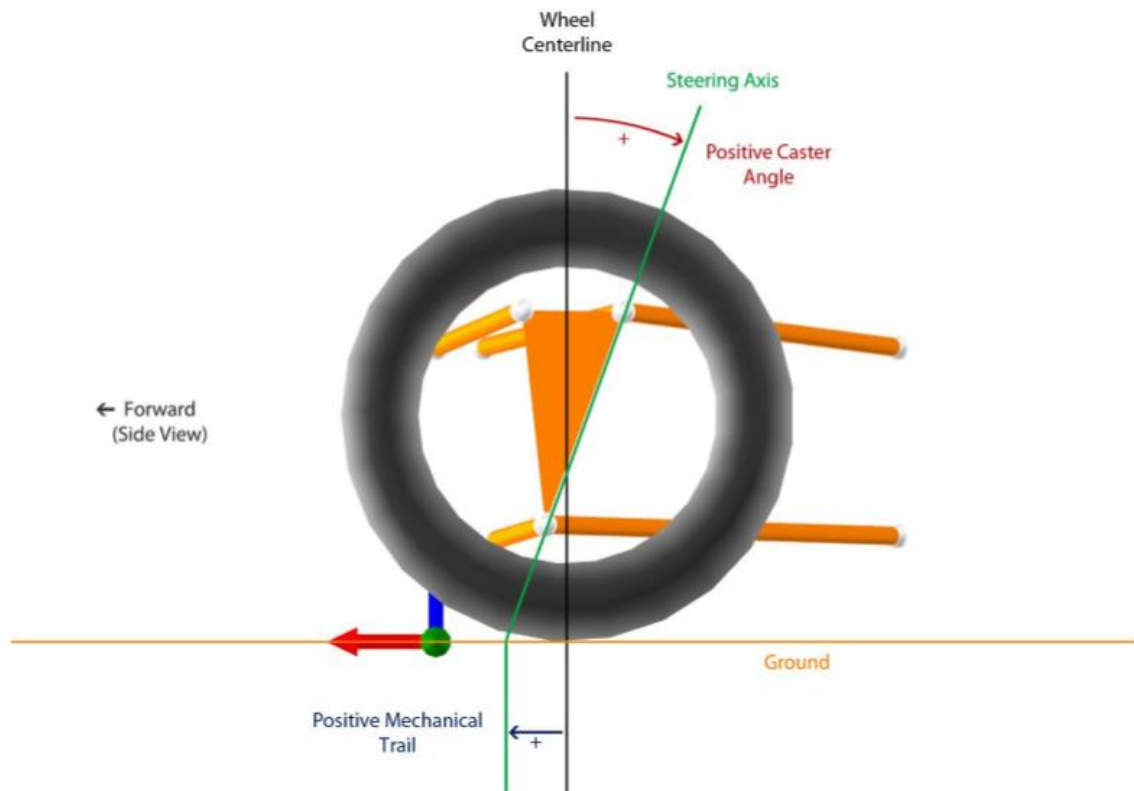


Figure 3.38: Mechanical Trail and Caster Angle representation in Optimum Kinematics

### King Pin Angle:

King pin angle is defined as the angle between the steering axis and an axis extending perpendicular from the contact patch, viewed front on (perpendicular to the vertical-lateral plane.)

Positive king pin angle is defined as the top of the steering axis being closer to the vehicle centreline.

### Scrub Radius:

The Scrub Radius is defined as the distance between the intersection of the steering axis and the ground measured to the center of the contact patch, viewed perpendicular to the vertical lateral plane.

Positive scrub radius is defined as the steering axis intersecting the ground plane between the vehicle centreline and the contact patch.

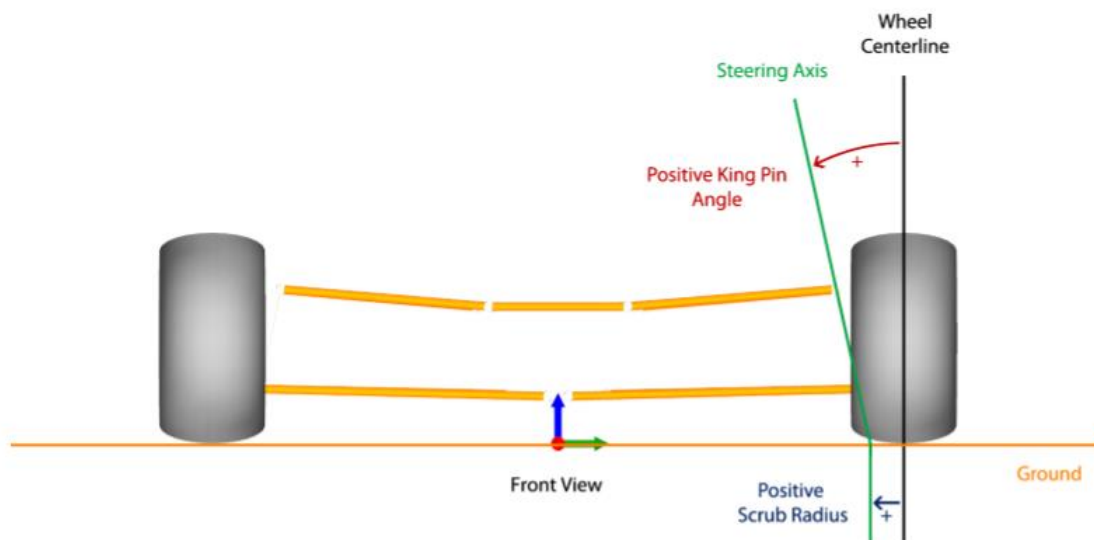


Figure 3.39: Scrub radius representation in Optimum Kinematics

#### Front View Instantaneous Center:

The front view instant center point is where the instant axis intersects a vertical plane between the two front or rear tire contact patches.

#### Front view Virtual Swing Arm length:

The front view virtual swing arm length is defined as the distance between the contact patch and the front view instant center.

#### Front view Virtual Swing Arm angle:

The front view virtual swing arm angle is defined as the angle formed between the ground plane and the front view virtual swing arm, viewed from the front. Positive angles occur when the instant center is located above the ground.

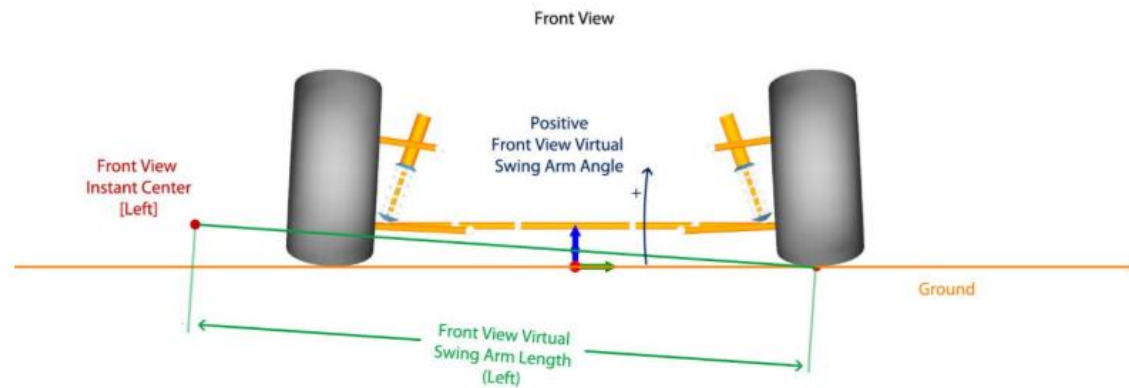


Figure 3.40: Front view virtual swing arm representation in Optimum Kinematics

#### Side view Instantaneous center:

The Side view instant center point is where the instant axis intersects a vertical plane between the two left or right contact patches.

#### Side view Virtual Swing Arm Length:

The side view virtual swing arm length is defined as the distance between the contact patch and the side view instant center.

### Side view Virtual Swing Arm angle:

The side view virtual swing arm angle is defined as the angle that the side view virtual swing arm forms with the ground plane, viewed from the side (longitudinal-vertical) plane.

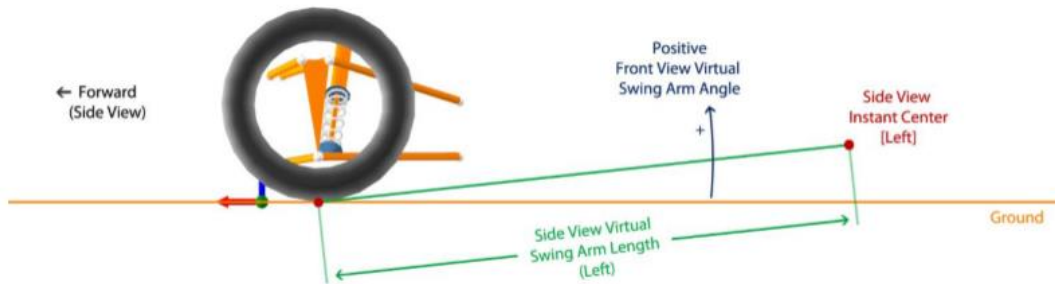


Figure 3.41: Side view Swing Arm length and angle

### Instantaneous Axis:

The instantaneous axis is the axis which the non-suspended mass rotates around as the suspension articulates. Each corner of the car has its own instantaneous axis. The instantaneous axis is located by moving the wheel up and down by a small increment and determining the point about which the wheel rotates, taking into account all suspension links –including tie rods and suspension.

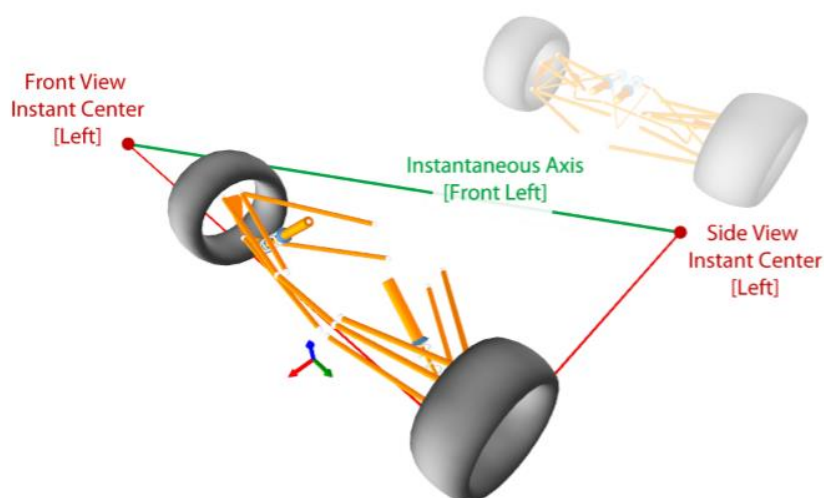


Figure 3.42: Representation of the instantaneous axis

### Kinematic Pitch Center:

Figure 3.43 shows the definition of the kinematic pitch center

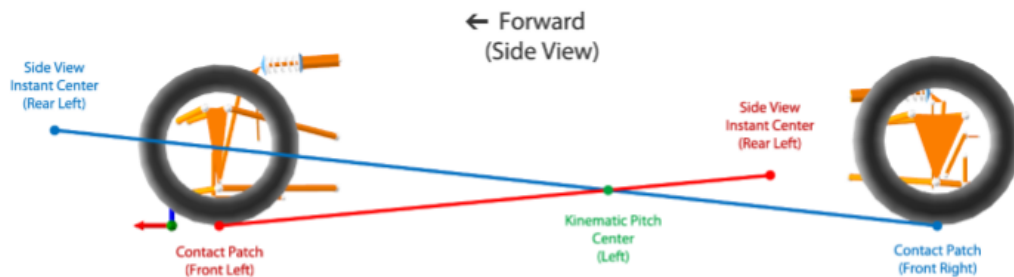


Figure 3.43: Representation of the kinematic pitch center in Optimum Kinematics

### Kinematic Pitch Axis:

Figure 3.44 shows the definition of the kinematic pitch axis.

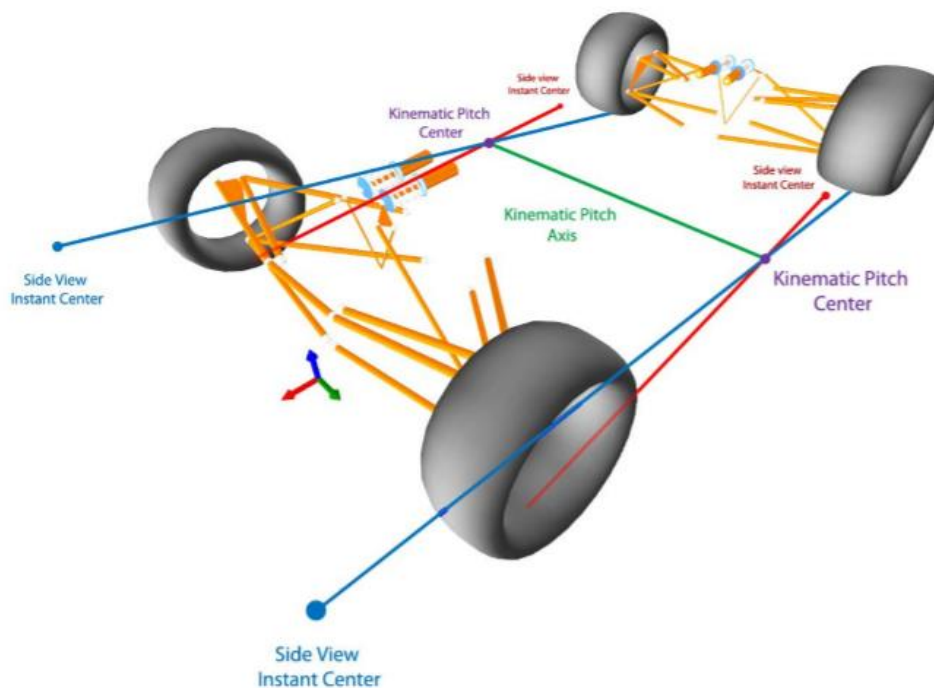


Figure 3.44: Representation of the kinematic pitch axis

### Kinematic Roll Center:

The Kinematic Roll Center is the intersection point between the left and right planes, which are defined by the instant axis and contact patch point, and the vertical plane between the left and right contact patches. This point is not stationary – it can move as the suspension articulates.

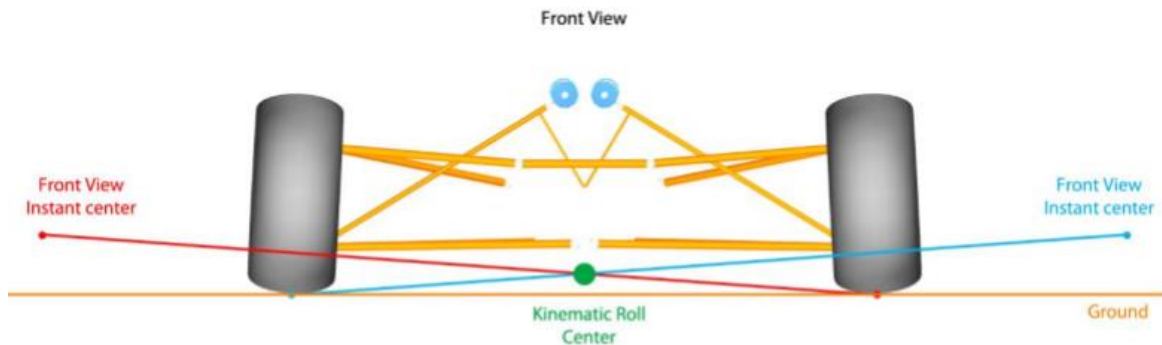


Figure 3.45: Definition of the roll center in Optimum Kinematics

### Kinematic Roll Axis:

The Roll Axis is a line drawn between the Front and Rear Roll Centers. This is the axis about which the suspended mass rotates around.

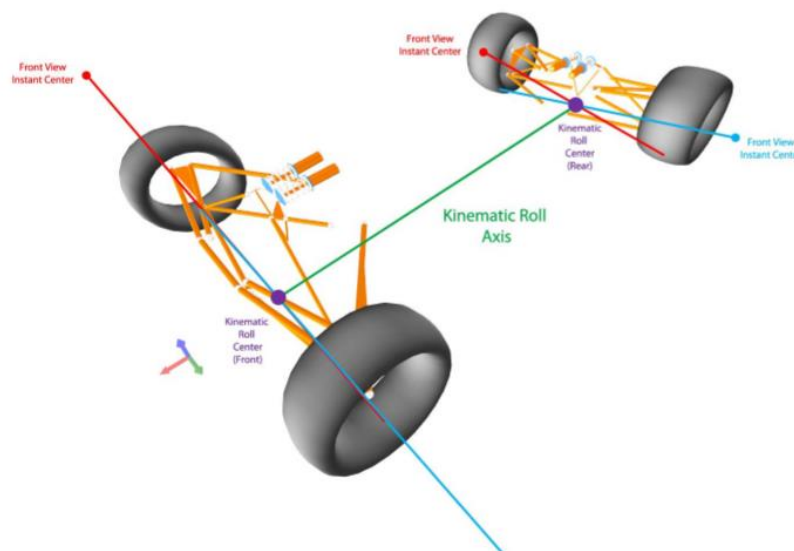


Figure 3.46: Representation of the axis' roll center

### Roll Axis Inclination:

The Roll Axis Inclination is defined as the angle between the kinematic roll axis and the ground plane, viewed perpendicular to the vertical-longitudinal plane.

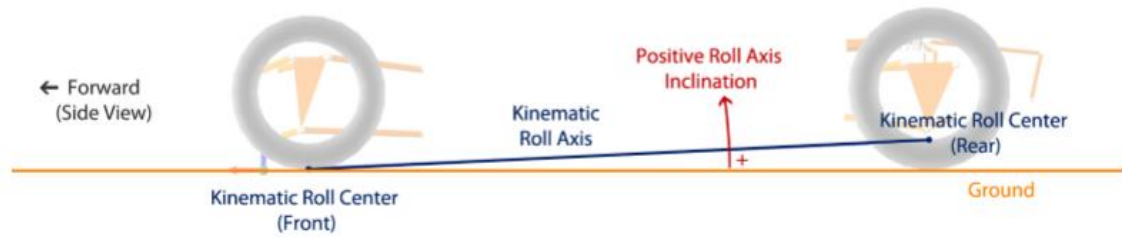


Figure 3.47: Representation of the roll axis' inclination

### Ackermann percentage definition:

The Ackerman percentage is defined as a function of the inside steer angle, the outside steer angle, the front track and the vehicle wheelbase. The equations are as follows.

$$Ackerman = \tan^{-1} \left( \frac{wheelbase}{\frac{wheelbase}{\tan \delta_{outside}} - track_{front}} \right)$$

$$Ackerman_{percent} = \frac{\delta_{inside}}{Ackerman} \times 100$$

(Eq. 1)

### 3.6.1. Base setup results

The graphics in this section show the data obtained from the analysis files generated after the different simulations of the 6 motions defined in section 3.5.2 combined with the starting point geometry of the suspension system defined in section 3.5.1.

#### Roll Motion Results:

The following graphics contain all the plotted channels after the car completed the roll simulation with the starting point suspension system configuration selected.

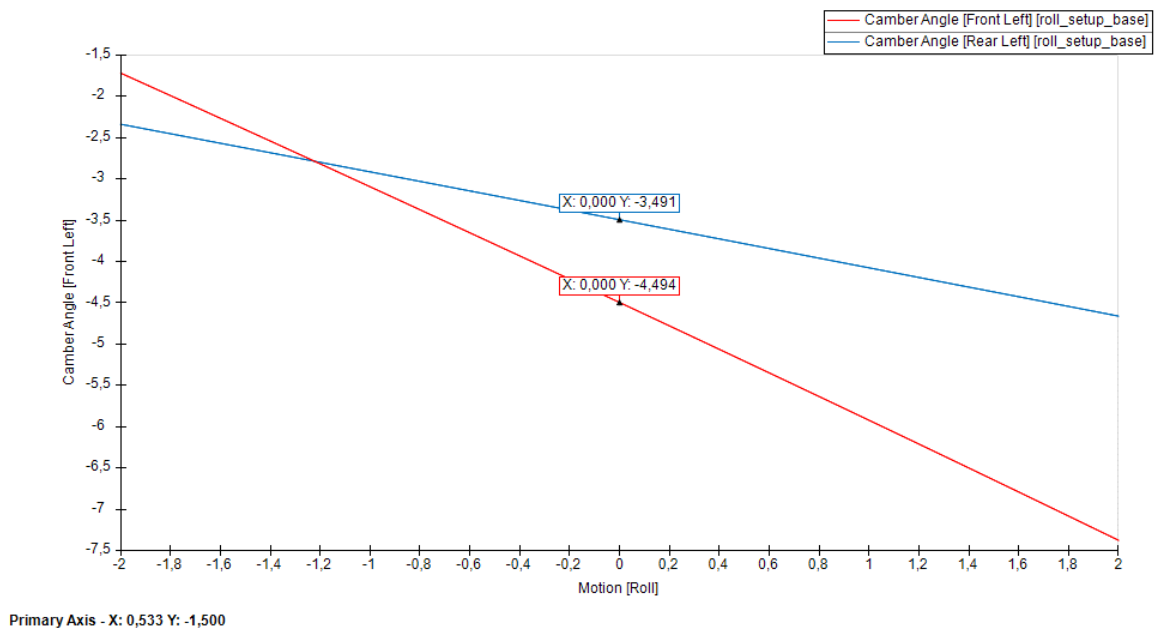


Figure 3.48: Camber angle vs Roll

In figure 3.52 it is important to check that in any roll angle value of the car, the geometry of the suspension grants that the camber angle never assumes a positive value, as that would produce jacking effect, thus drastically reducing the stability of the car during cornering.



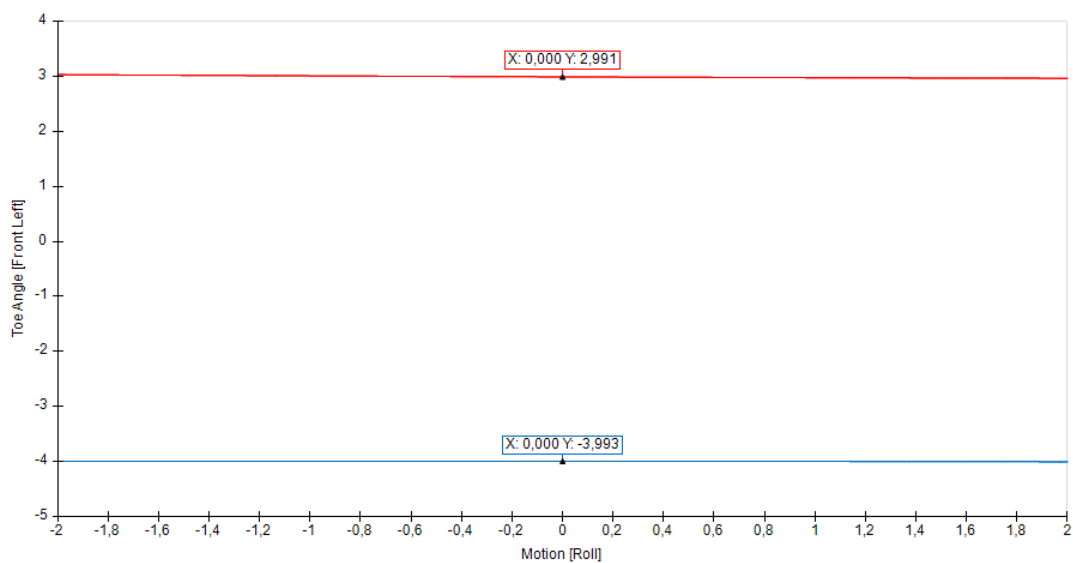


Figure 3.49: Toe angle vs Roll

Figure 3.53 is a useful graphic to determine if a vehicle suffers from Roll Steer. Depending on the suspension geometry, in a cornering event a car can have subtle variations of the toe angle of the car. If these values are too big, the car becomes unstable and its behaviour changes in corners, which makes it difficult to handle for the driver.

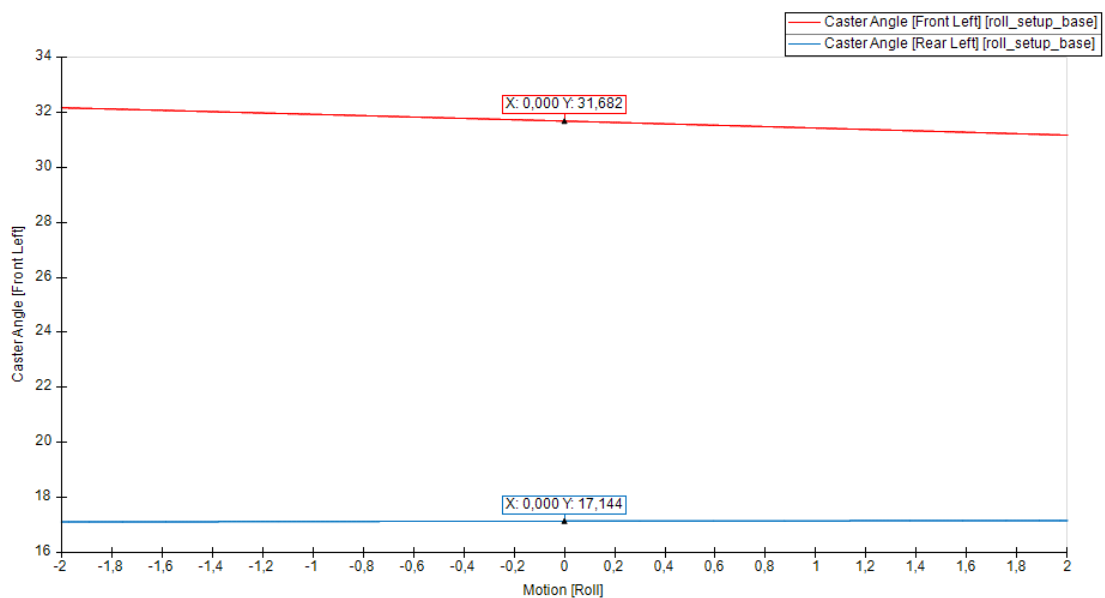


Figure 3.50: Caster angle vs Roll

In figure 3.54 it is important to ensure that the caster angle does not have extreme variations respective to the starting value, as it would mean that the car changes the feeling of the steering wheel and the resistance of the steering system in a cornering event. Thus, making the car more difficult to drive.

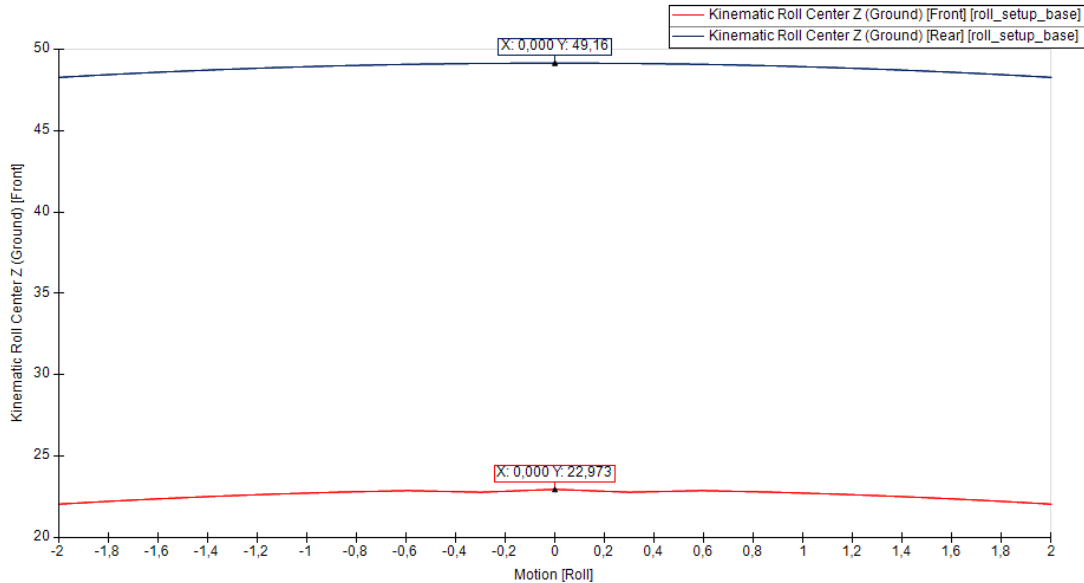


Figure 3.51: Kinematic Roll Center Z vs Roll

It is important to check in figure 3.55 that the rear roll center is always higher than the front roll center during all the roll motion.

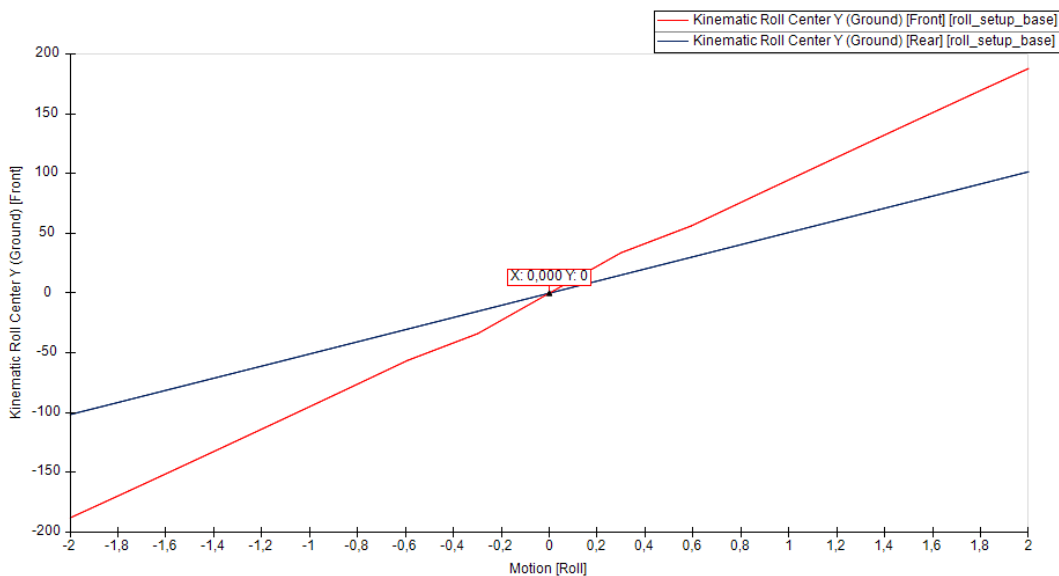


Figure 3.52: Kinematic Roll Center Y vs Roll

Figure 3.56 shows information about how much the roll center migrates from its center position in a pure cornering event. The greater the displacement (shown in mm), the more it is displaced. Theoretically this modifies the position of the roll axis of the car which worsens the car's behaviour in cornering if values are big enough.

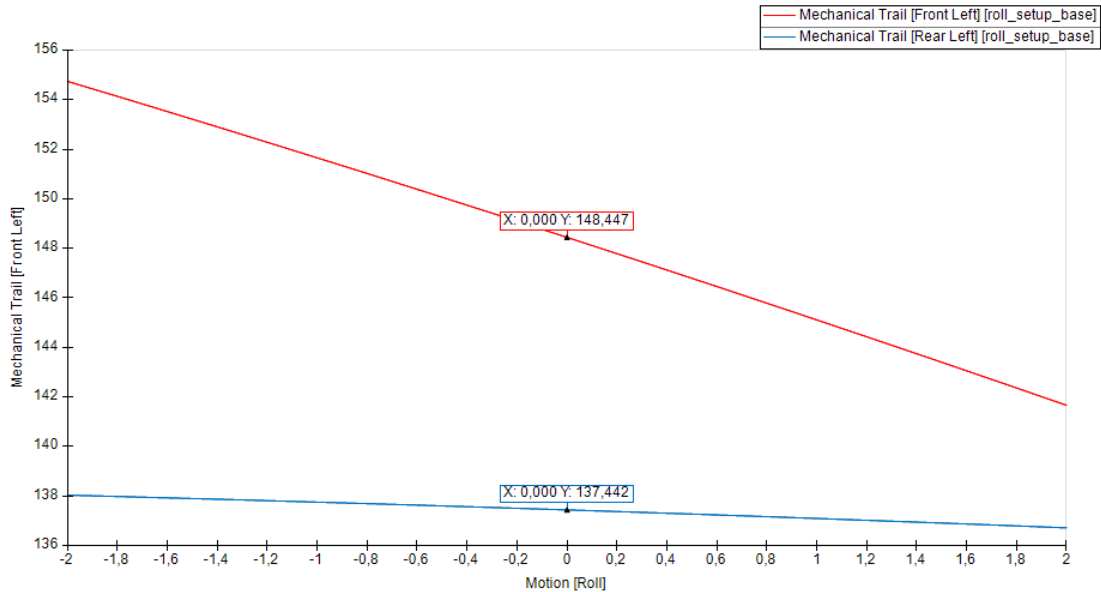


Figure 3.53: Mechanical Trail vs Roll

What it's important to observe in figure 3.57 here is that the mechanical trail value does not vary excessively during the roll motion and additionally that it does not assume negative values. That would make the feeling of the steering wheel to turn itself into the corner rather than having to apply a force to steer.

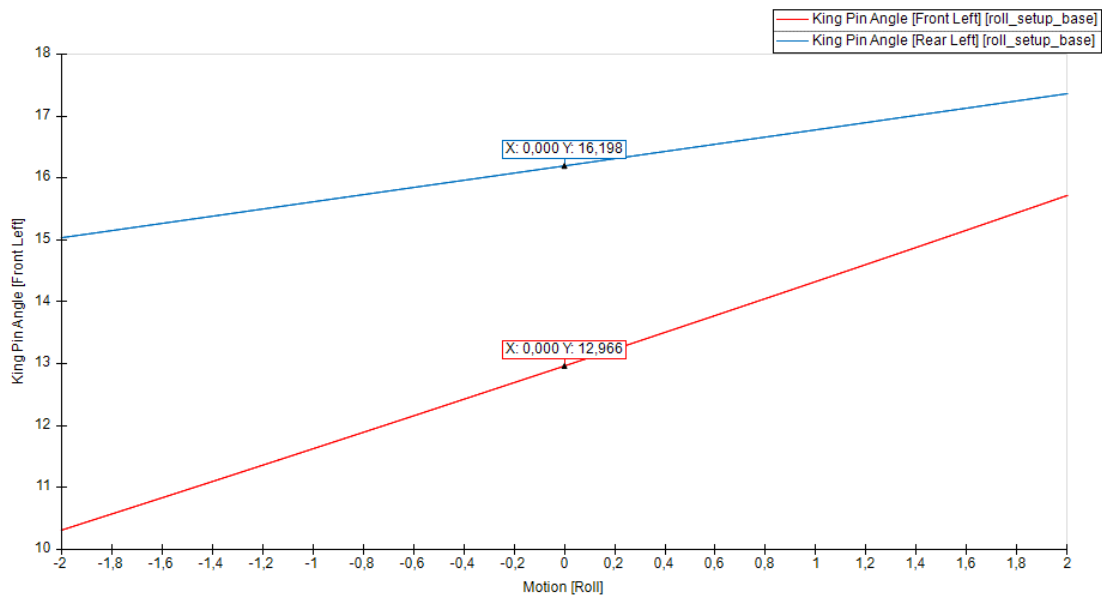


Figure 3.54: King Pin angle vs Roll

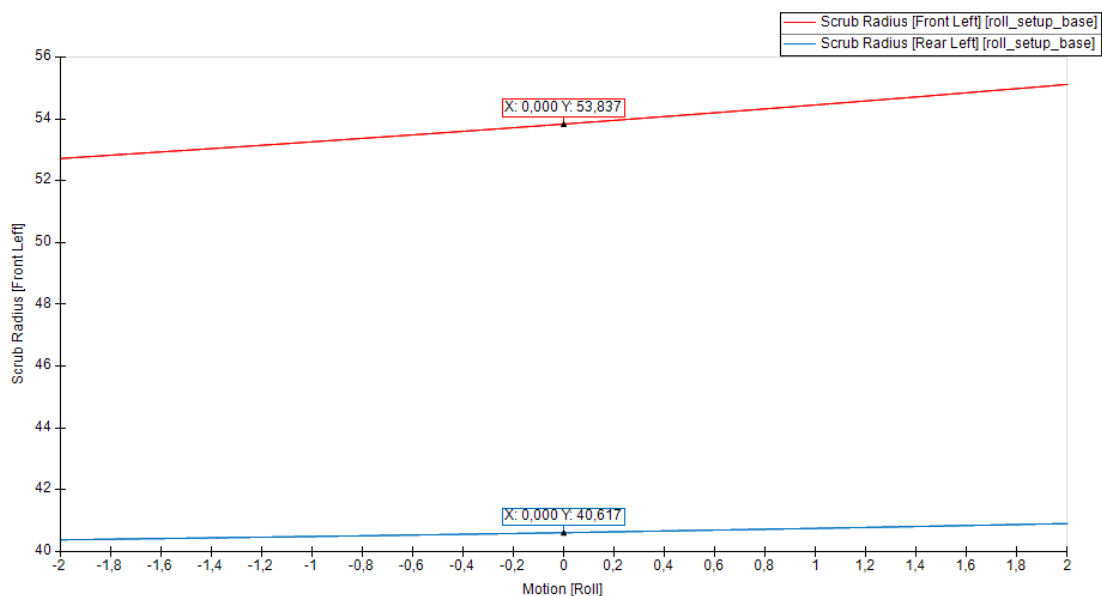


Figure 3.55: Scrub Radius vs Roll

It is important to check in figures 3.58 and 3.59 that the values don't vary excessively respect the original starting points during the cornering event as well as to check that the difference between the two values is in general terms similar.

### Heave Motion Results:

The following graphics contain all the plotted channels after the car completed the heave simulation with the starting point suspension system configuration selected.

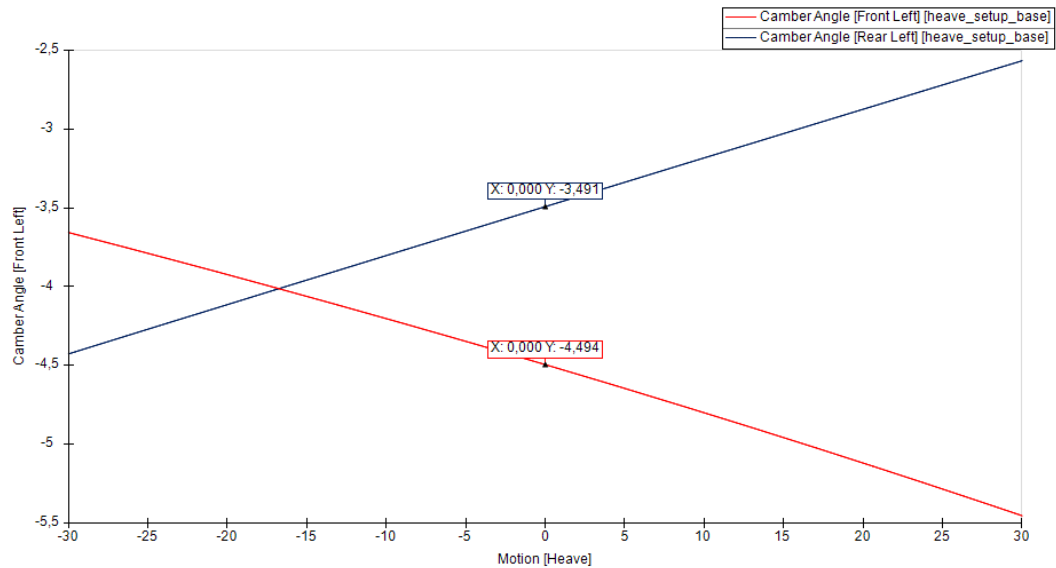


Figure 3.56: Camber angle vs Heave

Like the case of camber angle variation in roll motion, the important bit of figure 3.60 is to ensure that the camber angle never assumes a positive value in any moment of the bump event.

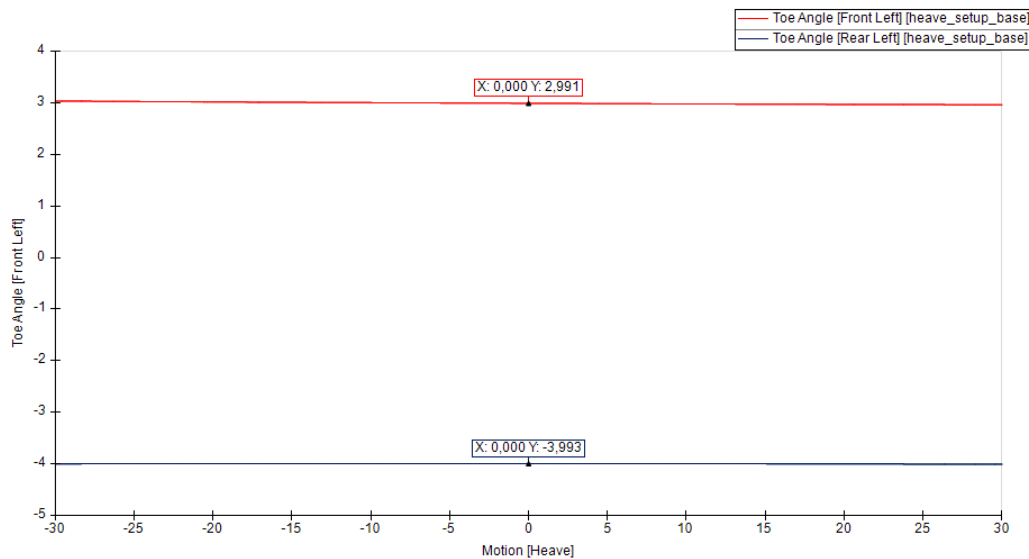


Figure 3.57: Toe angle vs Heave

Figure 3.61 reproduces the Bump Steer problem. Just like in the Roll Steer problem, depending on the geometry, a bump can alter the toe angle of a wheel, creating unstable behaviour of the car. Therefore, it is necessary to check that the variation is as little as possible.

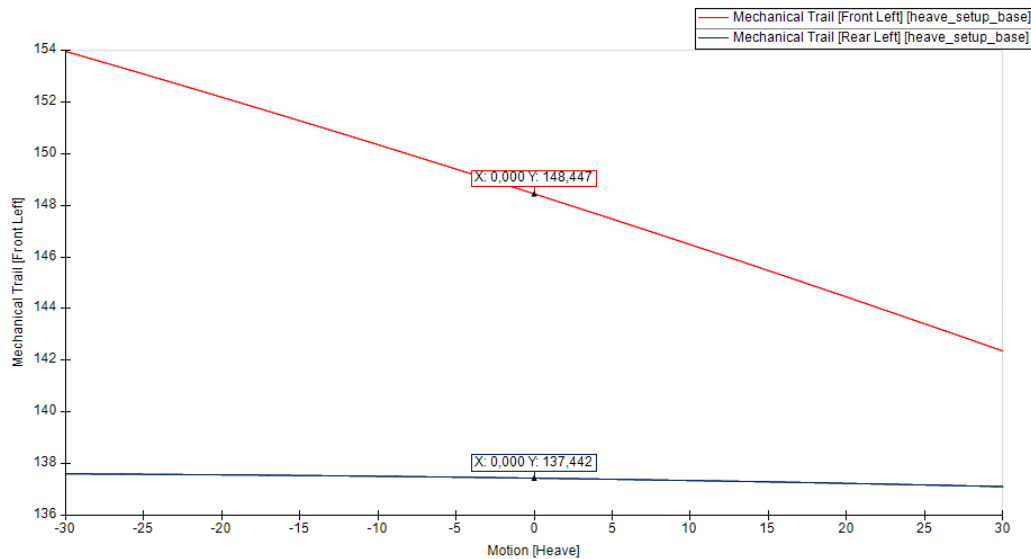


Figure 3.58: Mechanical Trail vs Heave

What it's important to observe in figure 3.62 is that the mechanical trail value does not vary excessively during the heave motion and additionally that it does not assume negative values. That would make the feeling of the steering wheel to turn itself into the corner rather than having to apply a force to steer.

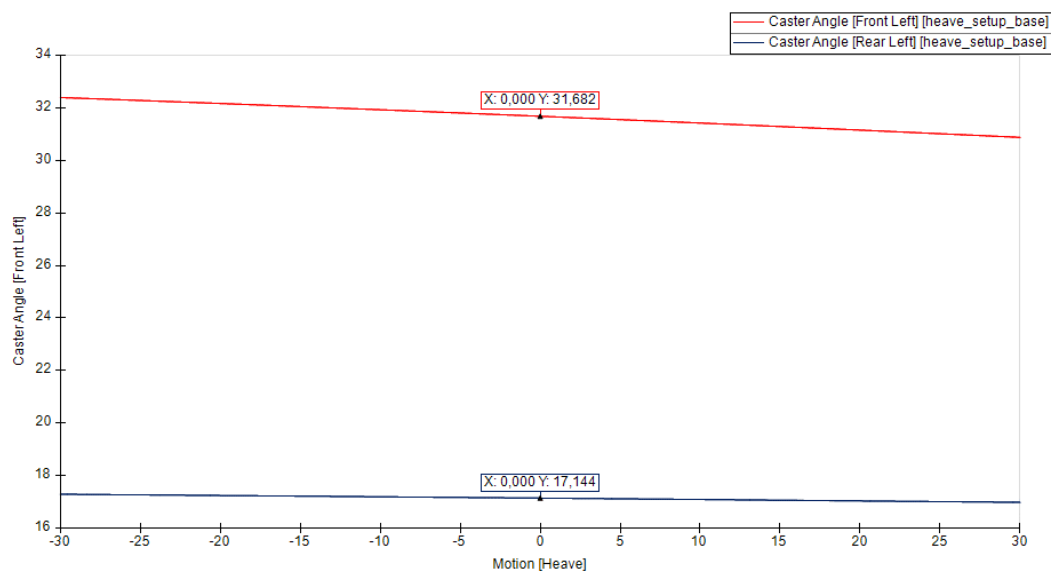


Figure 3.59: Caster Angle vs Heave

In figure 3.63 it is important to ensure that the caster angle does not have extreme variations respective to the starting value, as it would mean that the car changes the feeling of the steering wheel and the resistance of the steering system in a cornering event. Thus making the car more difficult to drive.

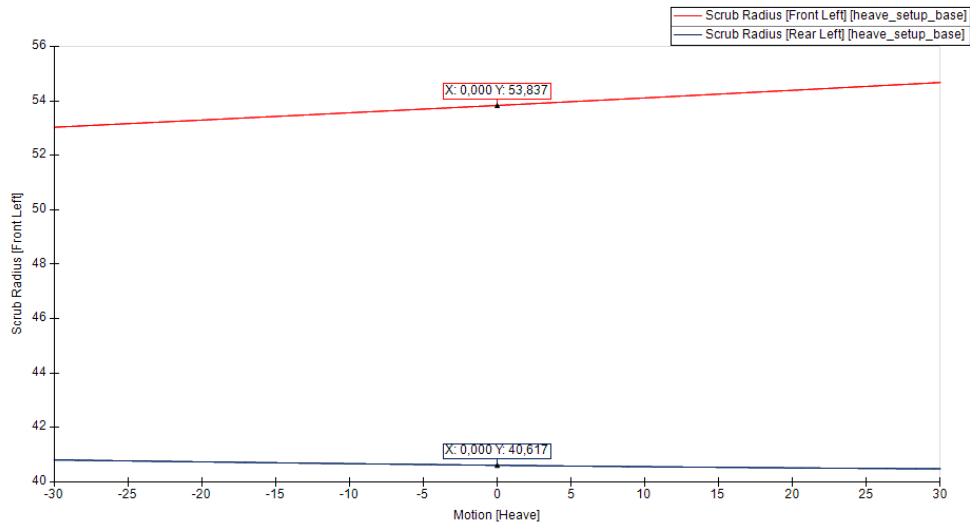


Figure 3.60: Scrub radius vs Heave

In figure 3.64 it is important to check that the values don't vary excessively respect the original starting points during the cornering event as well as to check that the difference between the two values is in general terms similar.

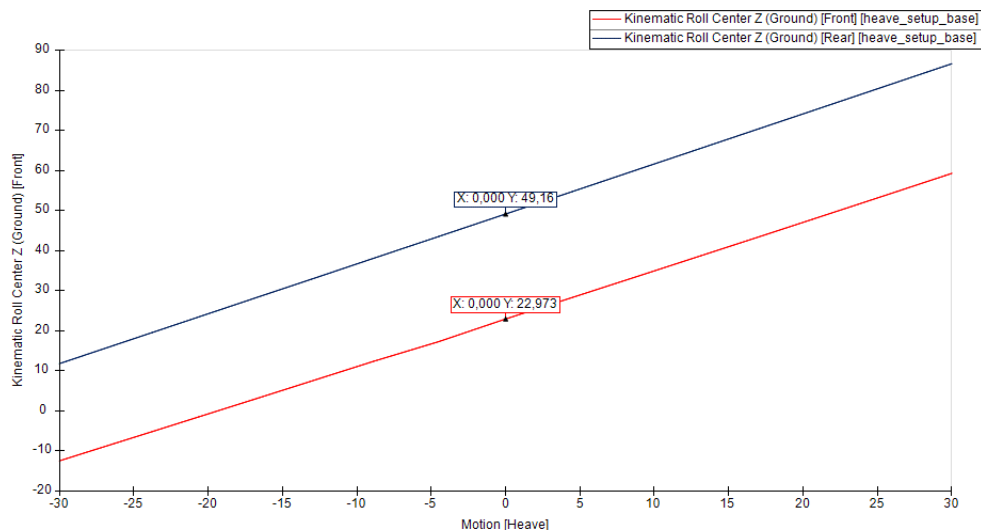


Figure 3.61: Kinematic Roll center Z vs Heave

In figure 3.65 it is important to check that the rear roll center is always higher than the front.

### Steering Motion Results:

The following graphics contain all the plotted channels after the car completed the steering simulation with the starting point suspension system configuration selected.

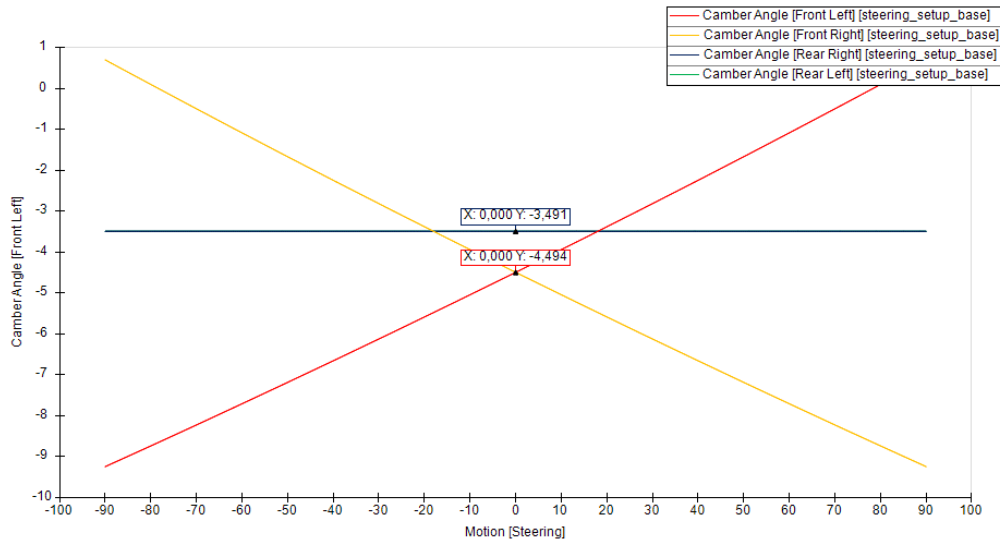


Figure 3.62: Camber angle vs Steering

In figure 3.66 it is important to check that all four wheels act properly, and to check that the camber angle does not assume positive values during the steering event. It is observable that in the baseline configuration the inside wheel loses camber and at maximum steering wheel input the value is slightly positive.

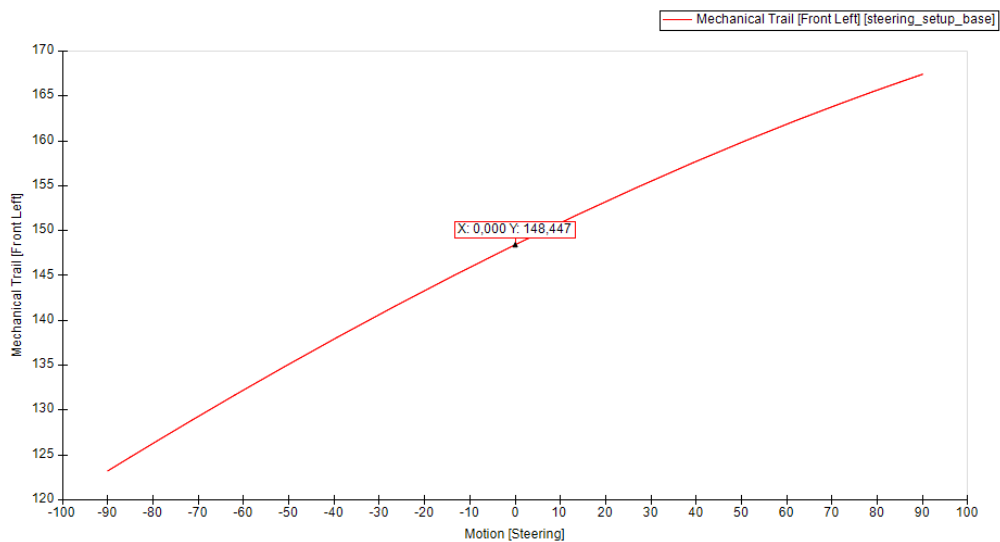


Figure 3.63: Mechanical Trail vs Steering



It is important to check in figure 3.67 that the Mechanical Trail doesn't assume negative values during the range of steering, or that would change the feel of the wheel, which would turn itself in the direction of the corner rather than having to apply a force to steer in that same direction. Additionally, it is important to check that the variation is somewhat uniform.

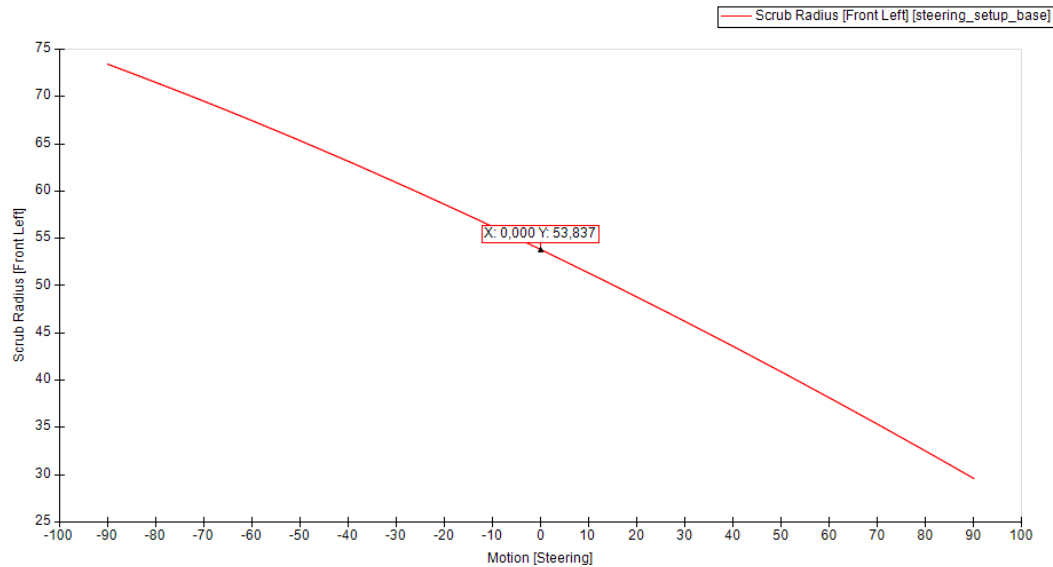


Figure 3.64: Scrub Radius vs Steering

In figure 3.68 it is important to check that the variation of the scrub radius during the steering event is linear, and that it does not assume negative values.

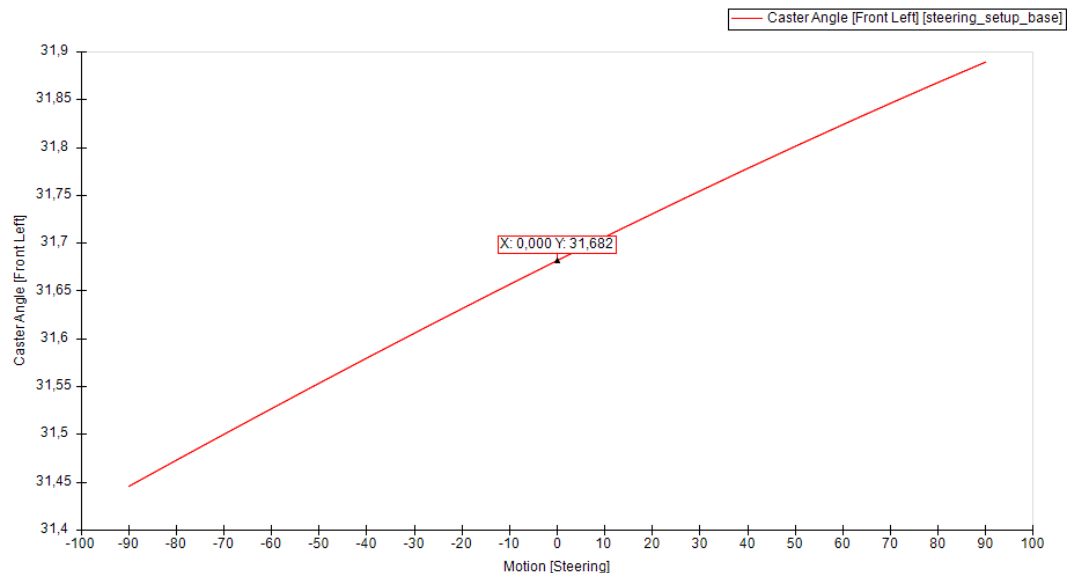


Figure 3.65: Caster angle vs Steering

### Pitch Motion Results:

The following graphics contain all the plotted channels after the car completed the pitch simulation with the starting point suspension system configuration selected.

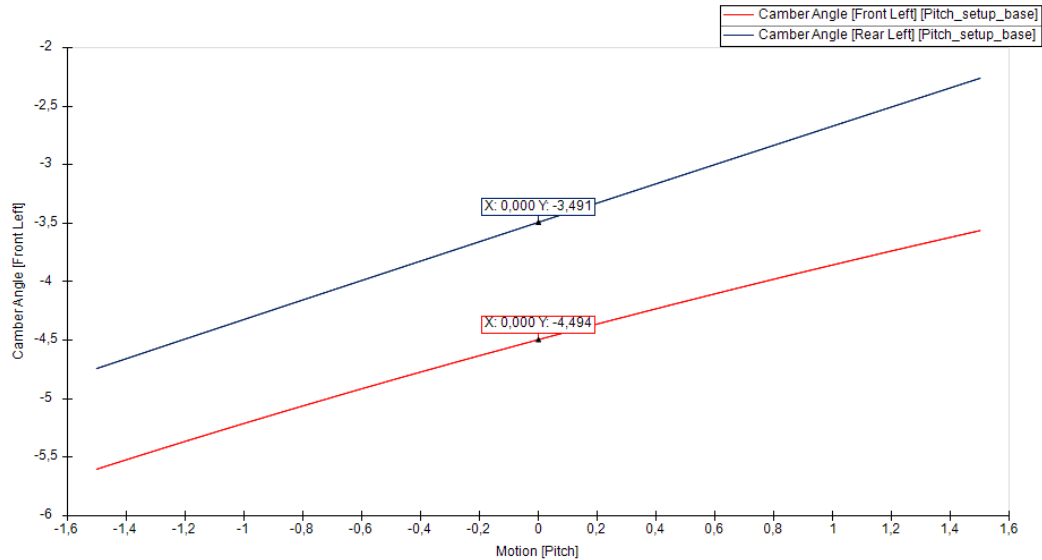


Figure 3.66: Camber angle vs Pitch

In figure 3.70 it is important to check that in any pitch angle value of the car, the geometry of the suspension grants that the camber angle never assumes a positive value, as that would produce jacking effect, thus drastically reducing the stability of the car during cornering.

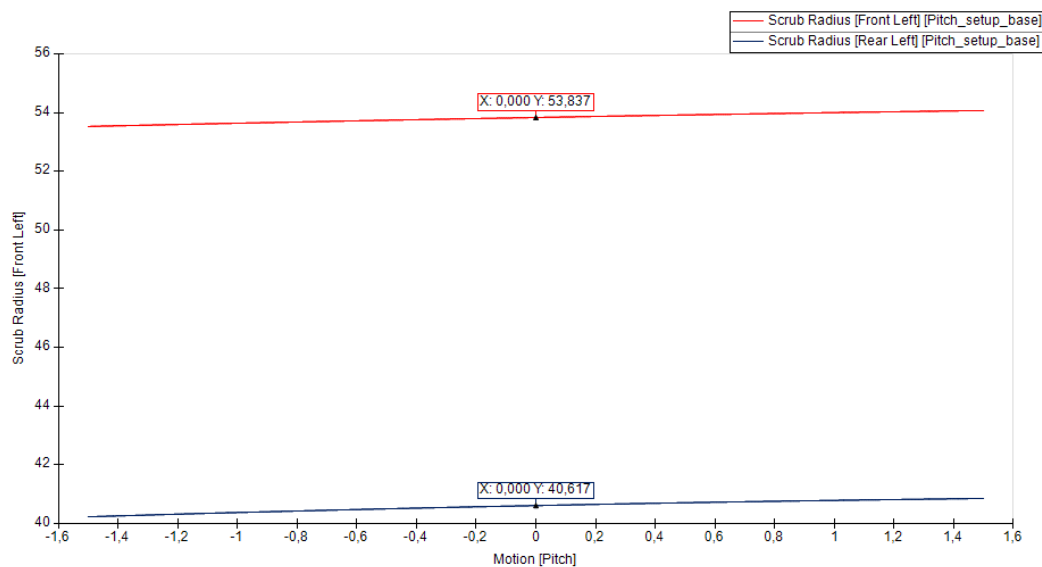


Figure 3.67: Scrub Radius vs Pitch

In figure 3.71 it is important to check that the values don't vary excessively respect the original starting points during the cornering event as well as to check that the difference between the two values is in general terms similar.

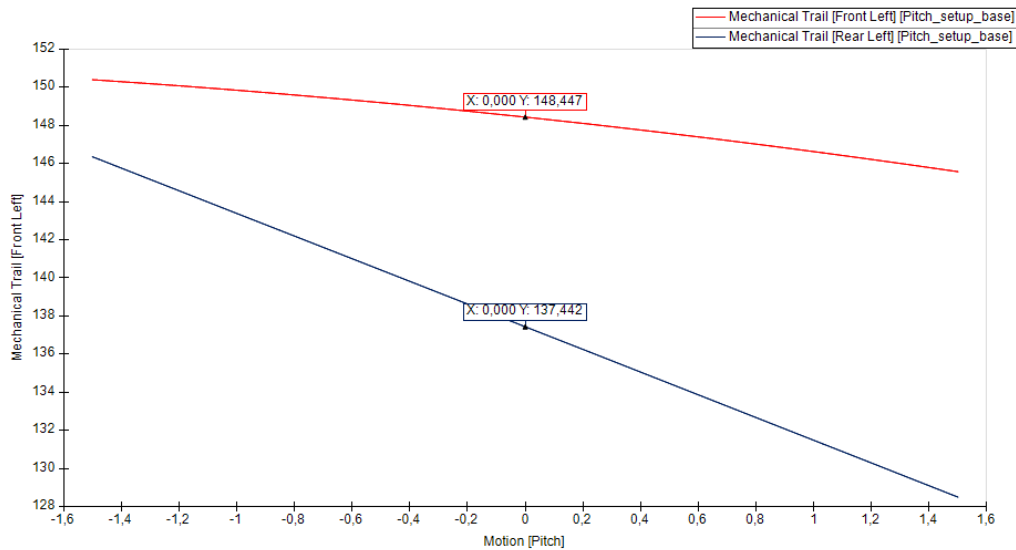


Figure 3.68: Mechanical Trail vs Pitch

It is important to check in figure 3.72 that the Mechanical Trail doesn't assume negative values during the pitch motion, or that would change the feel of the wheel, which would turn itself in the direction of the corner rather than having to apply a force to steer in that same direction. Additionally, it is important to check that the variation is somewhat uniform.

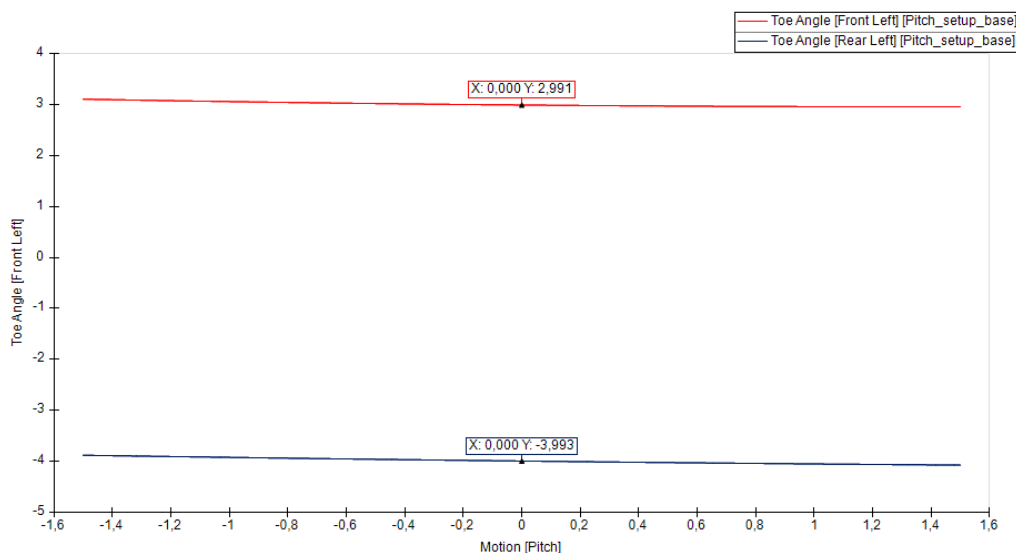


Figure 3.69: Toe angle vs Pitch

Just like in the Bump Steer and Roll Steer cases, in figure 3.73 it is important to check that the toe angle variation is minimal to ensure that the wheels won't change orientation during traction and breaking events.

### Corner Entry Results:

The camber of the four wheels is analysed during the whole manoeuvre to ensure no positive values are assumed. It is observable that the rear wheels experiment a bigger variation than the front ones, due to the geometry of the configuration of the real axle suspension system.

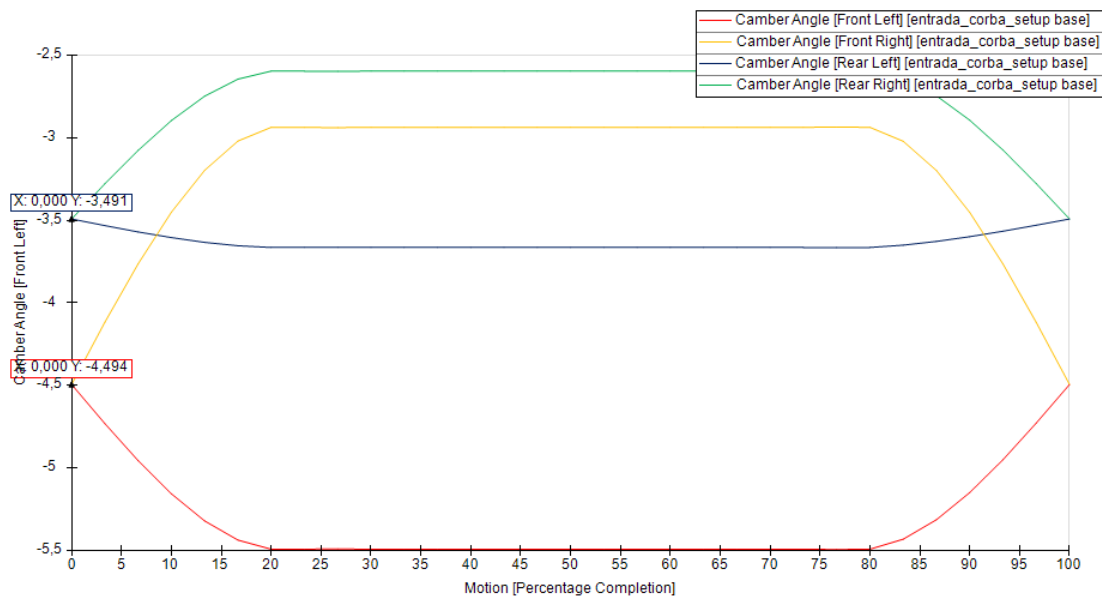


Figure 3.70: Camber angle vs motion completion in corner entry

### Corner Exit Results:

Like the corner entry event, the four camber angles of the wheels are analysed. It is observable that the rear wheels experiment a bigger variation than the front ones.

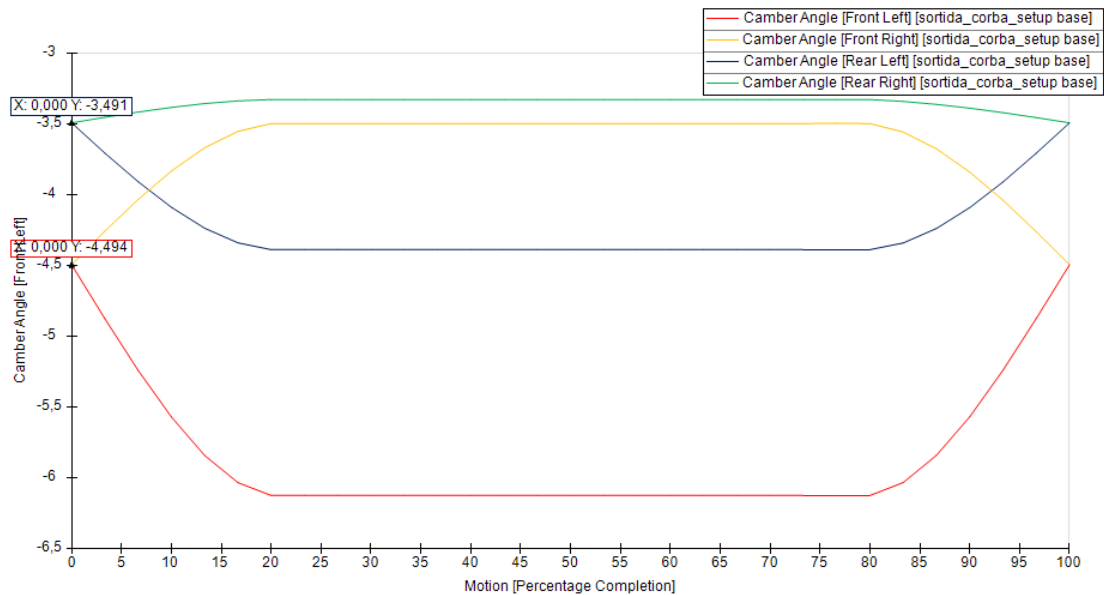


Figure 3.71: Camber angle vs motion completion in corner exit

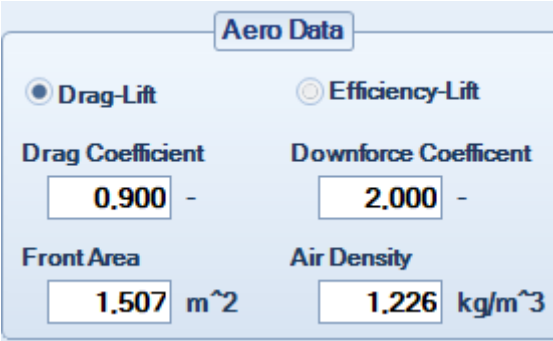
### 3.6.2. Base setup performance analysis

Once the design, analysis and results are finished in Optimum Kinematics the base configuration can be exported to Optimum Lap in order to virtually test the results in a real track scenario.

Adding to the suspension system and tyre data previously added, aerodynamic data, an engine, and a transmission must be added to create the full virtual vehicle. Additionally, the track information needs to be added in the software's required format.

Aerodynamic Data:

The aerodynamic data of the car is introduced using the menu the following figure shows.



The screenshot shows the 'Aero Data' menu in Optimum Lap. It features two radio buttons at the top: 'Drag-Lift' (selected) and 'Efficiency-Lift'. Below these are four input fields arranged in a 2x2 grid. The top-left field is 'Drag Coefficient' with a value of '0.900'. The top-right field is 'Downforce Coefficient' with a value of '2.000'. The bottom-left field is 'Front Area' with a value of '1.507' and a unit of 'm^2'. The bottom-right field is 'Air Density' with a value of '1.226' and a unit of 'kg/m^3'.

Parameter	Value	Unit
Drag Coefficient	0.900	-
Downforce Coefficient	2.000	-
Front Area	1.507	m <sup>2</sup>
Air Density	1.226	kg/m <sup>3</sup>

Figure 3.72: Aero data menu in Optimum Lap

The frontal area of the car is calculated using the technical specifications sheet (Section 3.2) and the air density is a standard reference value. Due to the lack of data for both the Drag and Downforce Coefficients, general single-seater values were grabbed from the Optimum Lap vehicle database.

These values remain untouched during all of the modifications to the configuration, as the study is focused in the suspension system only and the differences in lap times aim to be quantified only by suspension system configuration changes.

### Engine Data:

As stated in Section 3.2 – Table 3.2, the regulations of the Euroformula Open restricts the engine to a single-make Toyota 3SGE engine. As it is a commercial engine, a dyno data chart analysis is obtained to create the Engine Speed vs Engine Torque table shown below.



ENGINE DATA	
Add / Remove Torque Data	
	
Engine Speed (rpm)	Engine Torque (N.m)
2000	61.01
3000	101.69
4000	149.14
5000	220.99
6000	277.94
7000	338.95
7500	352.51
8000	345.73

Table 3.10: Engine Speed vs Engine Torque values in Optimum Lap's input data box

The graphical representation of the table is as follows:

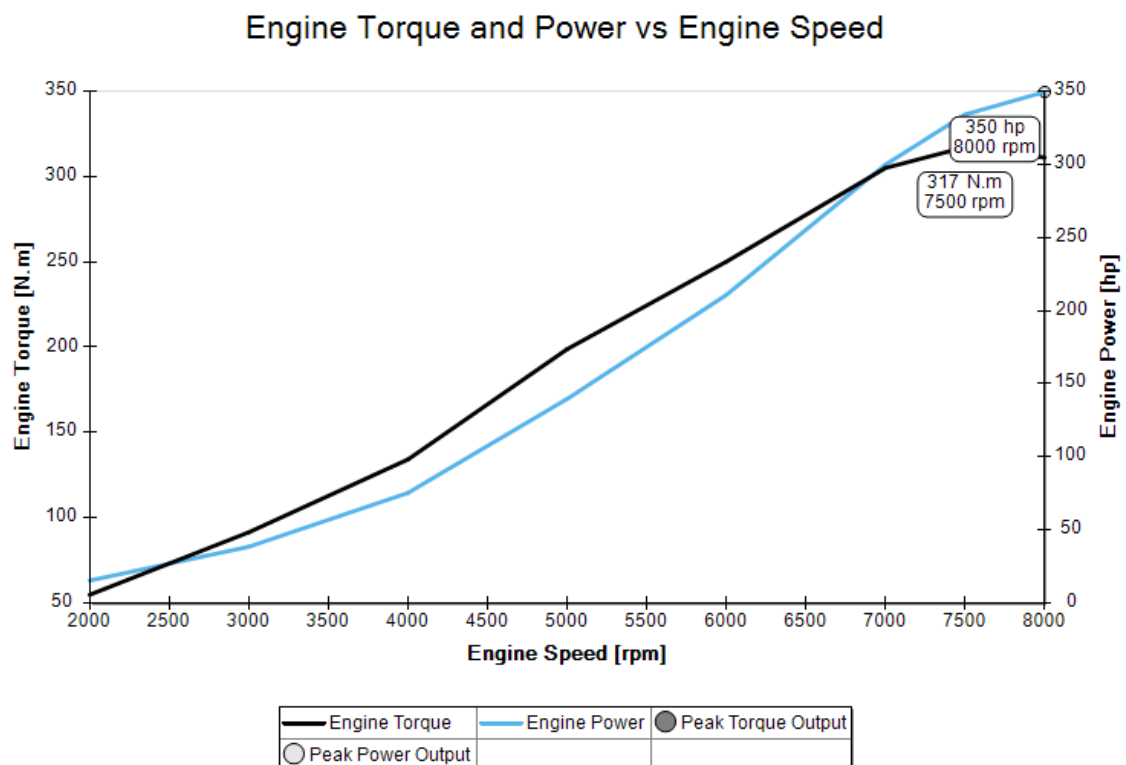
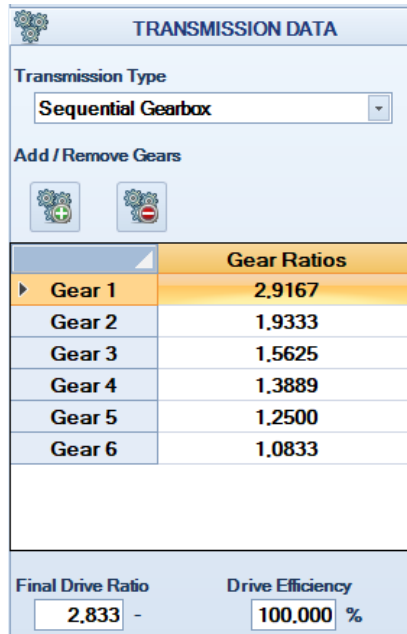


Figure 3.73: Engine Torque vs Engine Speed represented in Optimum Lap

### Transmission Data:

All the different gear ratios are extracted from the vehicle's user guide. As there is a list of recommended gear ratios dependent on the track, the standard values are chosen, as these values will remain unchanged during the different setup combinations in order to neglect the influence of different gear ratios combinations on the performance of the car.



TRANSMISSION DATA	
Transmission Type Sequential Gearbox	
Add / Remove Gears [Add Gear] [Remove Gear]	
	Gear Ratios
► Gear 1	2.9167
Gear 2	1.9333
Gear 3	1.5625
Gear 4	1.3889
Gear 5	1.2500
Gear 6	1.0833
Final Drive Ratio: 2.833 - Drive Efficiency: 100.000 %	

Table 3.11: Gear Ratio values in Optimum Lap



### Vehicle Traction Model:

Once all the data is introduced a general overview of the traction of the car can be viewed in order to preview how the car performs according to the data introduced above.

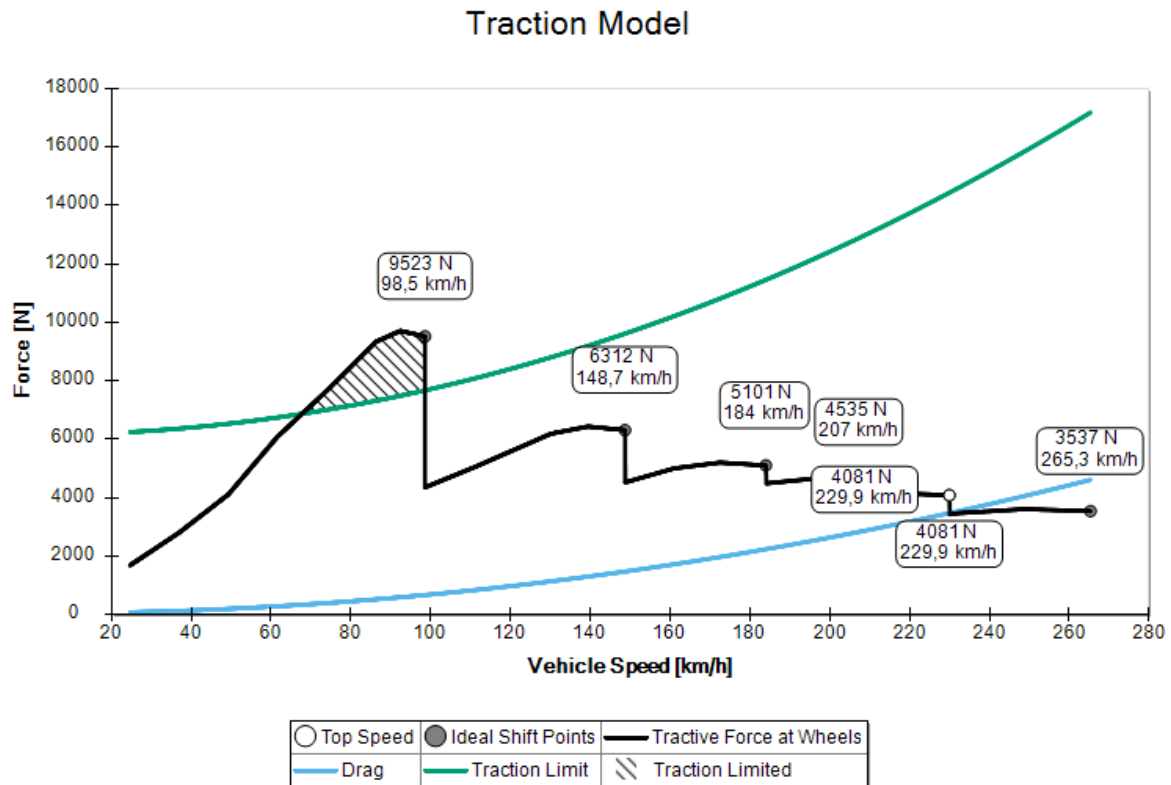


Figure 3.74: Vehicle traction model in Optimum Kinematics

It is observable that the car has wheel spin during first gear, something that occurs to this vehicle in real life during first gear.

### Track definition:

Once the required systems for the vehicle are defined in Optimum Lap, the track chosen for the study needs to be defined. Said chosen track is the Circuit of Spa-Francorchamps, in Belgium.

This track is part of the calendar of the Euroformula Open championship and it is regarded as one of the most complete tracks in the calendar.

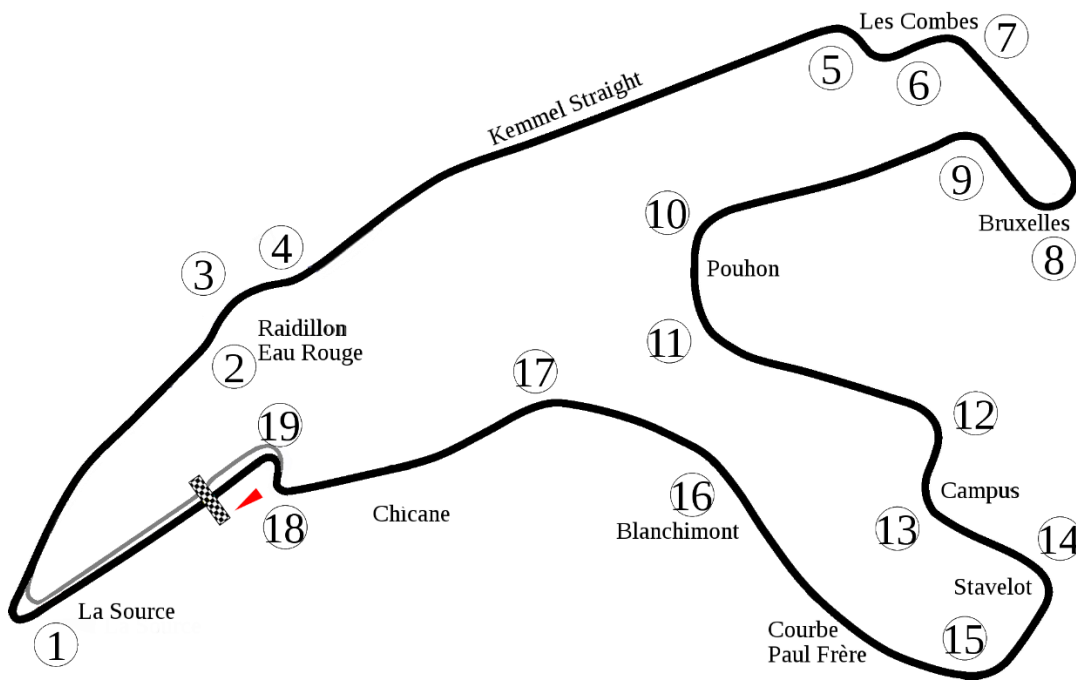


Figure 3.75: The circuit of Spa-Francorchamps

As figure 3.81 shows, the track combines fast sections as in Blanchimont with slow corners as La Source. The changes in elevation in the Eau Rouge section, as well as in the Campus makes for a good testing scenario for the car and its suspension system.

The track's specifications for the software are found in Optimum Lap track database and they are introduced like figure 3.82 shows:

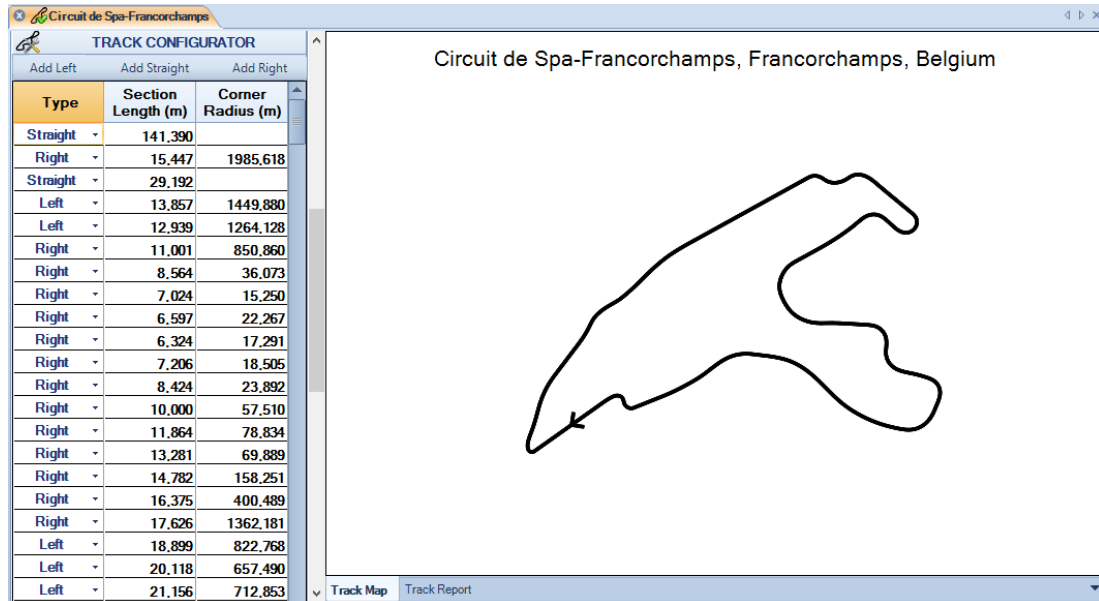


Figure 3.76: Spa-Francorchamps represented in Optimum Lap

Once the configuration of the car has been completed and the track is defined, the simulation can be performed.

Results comparing the starting point configuration and the custom configuration of the suspension system are shown in a later section.

### 3.7. Obtaining Results

The analysis of the custom configuration of the suspension system is done in this section, together with a comparison against the base configuration setup previously analysed in section 3.6.

The chosen setup is found iterating different combinations of the available options to modify the suspension system kinematics the Dallara F.317 offers. These options are explained below.

### 3.7.1. Setup options

The different possibilities to modify the kinematic configuration of the suspension system of the Dallara F-317 are listed below in the same style as the manufacturer exposes them in the user's guide.

- Rear Suspension pickup points
- Pushrod Adjuster
- Toe Adjuster
- Camber Adjuster
- Caster Adjuster
- Front Roll Center Height
- Pushrod Longitudinal Position

General comments from the manufacturer about the car's setup:

- In fast corners aerodynamics (ride heights and wing settings) have more influence on the balance than in slow corners.
- The weight distribution is important in slow and fast corners and together with the differential settings there are the most important contributors to the mechanical balance of the car.
- Dampers must be tuned to the chosen springs and not vice versa.
- Cold race tyres won't be able to generate the required grip. No car can reach its limit on too cold tyres, regardless of being balanced or not.
- General rule. It is recommendable to run the car as low as possible.

It is worth noting that the following rules apply during the entirety of the setup section:

A positive change in:

- Height: Car Rises
- Toe: Toe-Out
- Camber: Upper part of rim upward
- Castor: Lower part of the upright points ahead

### 3.7.1.1. Rear suspension pickup points

The following table lists the different rear suspension pickup point possibilities and the next figure shows the corresponding combinations of the table represented in the car. It is important to note that each point changed in Figure 3.84 equals  $\pm 10$  mm of change in coordinate Z.

CFG	Roll centre height /Altura C. Balanceo	Camber change / Variación Caída	Anti-rise / Anti-Levantamiento	Anti-squat / Anti-hundimiento	To adjust 'caster' adjust joint / Para ajustar avance ajusta	Spring Motion Ratio / Relación desplaz. muelle
	@ static ride height / @ altura estática	for 10mm travel / Para 10 mm de recorrido	%	%	+ means longer / +significa más largo	wheel/spring Rueda / Muelle
A-1	std	18'	7	49	-	1.289
B-2	-19	14'	7	49	-	1.245
C-1	18	22'	7	49	-	1.328
D-1	-1	20'	68	71	-0.5 turns	1.357
E-2	-19	15'	68	71	-0.5 turns	1.310
F-1	7	20'	22	38	-0.5 turns	1.321
G-2	-12	16'	22	38	-0.5 turns	1.276
H-2	-5	18'	37	27	-1.5 turns	1.308
I-1	12	21'	-9	61	+1 turn	1.296
L-1	-7	17'	-9	61	+0.5 turns	1.257
M-1	4	20'	38	92	+1 turn	1.330
N-1	14	22'	37	27	-1.5 turns	
O-1	-5	17'	37	27	-1.5 turns	
P-1	31	27'	37	27	-1.5 turns	
Q-1	2	19'	52	16	-2.5 turns	
R-1	20	24'	52	16	-2 turns	

Table 3.12: Rear suspension adjustment possibilities

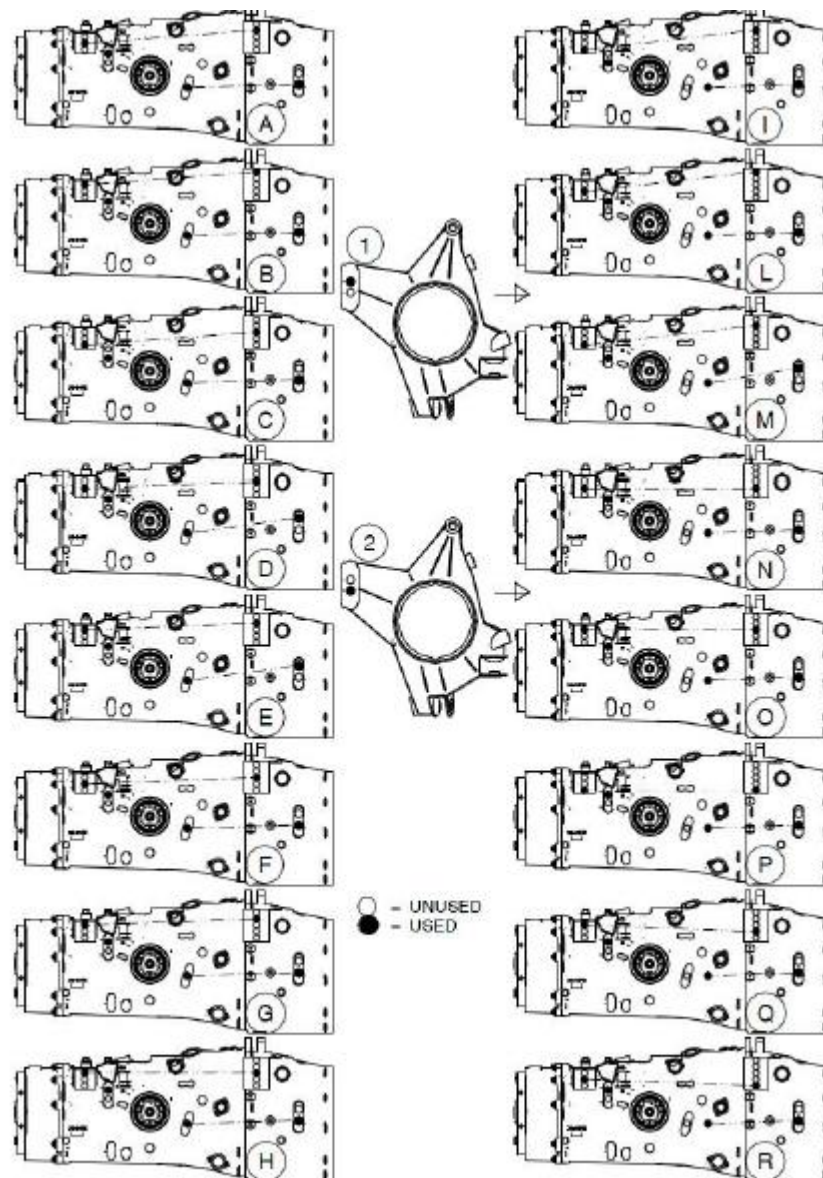


Figure 3.77: Rear suspension pickup points options

### 3.7.1.2. Pushrod adjuster

- Front Axle, 1 complete turn of the rod:
  - + 5,663 mm of height
  - + 10'30" of camber
- Rear Axle, 1 complete turn:
  - + 6,082 mm of height
  - + 11'17" of camber

### 3.7.1.3. Toe adjuster

- Front Axle, 1 complete turn of the rod:
  - + 38'42" of toe
- Rear Axle, 1 complete turn of the rod:
  - + 3,33 mm of height
  - + 19' of camber
  - - 46'30" of toe

### 3.7.1.4. Camber spacer

- Front Axle, 1 complete turn of the rod:
  - + 26'46" of camber
  - + 0,27 mm of height
- Rear Axle, 1 complete turn of the rod:
  - + 23'20" of camber
  - + 1,98 mm of height

### 3.7.1.5. Caster adjuster

- Front Axle, 1 degree of change:
  - - 0,86 mm of height
  - - 10'5" of camber
  - + 7'26" of toe
- Rear Axle, 1 degree of change:
  - - 1,55 mm of height
  - + 1'30" of camber
  - - 1'44" of toe

### 3.7.1.6. Front roll center height

Front Roll Center height can be changed by moving the spacer relative to the wishbone spherical joint. When 1 step is changed to a higher roll center, the pushrod length has to be shortened by 1/12 register turns to keep the same front height. When adjusting the RC height camber gain versus wheel travel varies a little.

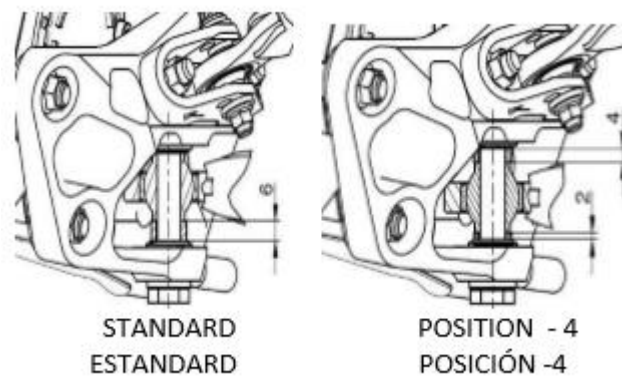


Figure 3.78: Roll center bearing adjustment

Configuration Standard -> RC comes from Optimum

Configuration -2mm -> RC height +9,4 mm / Camber Change 16'

Configuration -4mm -> RC Height +18,5 mm / Camber Change 15'

Configuration -6 mm -> RC Height +27,2 mm / Camber Change 14'



### 3.7.1.7. Pushrod longitudinal position

The F317 inherits the F312's Upright Mounted Pushrod type front suspension. Pushrod position is adjustable in longitudinal sense. Extra Load is transferred to the corner front inner wheel, potentially reducing under-steer thanks to a more equal vertical load between front inner and outer wheels. Worth noting that at higher values the steering force increases. Load transfer at the rear axle increases accordingly, but in the opposite direction, the inner wheel gets unloaded.

Point 2 is the reference point.

Point 1 -> -72% transfer load

Point 3 -> +72% transfer load

Point 4 -> +144% transfer load

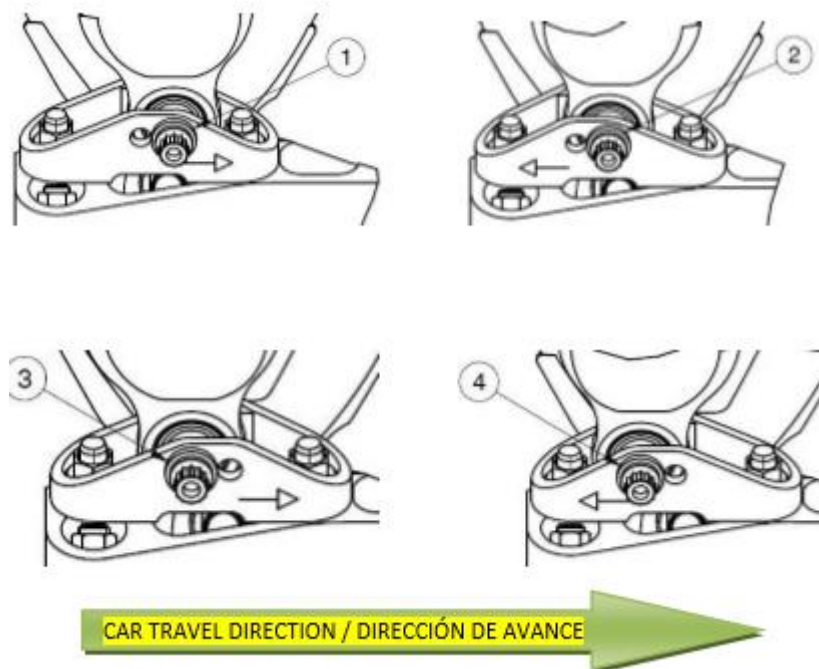


Figure 3.79: Different points for pushrod longitudinal adjustment

### 3.7.2. Summary of the alternatives tested

Due to the iteration process, the different possibilities explained above were tested to evaluate their respective influence and impact in performance.

Several conclusions while testing the different options are explained below. Those were the factors that led to the election of the chosen solution.

- Modifying the ride height of the car is a factor that has a big impact in performance. Lowering the height of the car has proven to be a determinant factor in performance for the chosen track, as it improves very significantly the performance of the car in fast corners.
- Modifying the camber variations during cornering events and static camber is a factor with important impact in performance. As it maximizes tyre contact to the ground and increases grip, helping significantly during slow corners.
- Caster variation has moderate influence on car performance. Straying too far from the base setup values increases the required cornering force, diminishing performance.
- The roll center variations of the car didn't offer significant variations in performance and in all options the focus was to ensure that the difference between front and rear roll centers remained similar.
- Toe variation has, generally a bad performance impact, as the starting values are already high. Further decreasing it had no significant impact on performance if the values are kept similar to the base values.

#### ALTERNATIVE CHOSEN:

The chosen alternative found that tries to maximize the above-mentioned statements uses the following options:

- The E-2 configuration for the rear suspension pickup points was chosen. The height and camber change that derive from the change has been adjusted accordingly.
- Adjusted the front pushrod of the car to keep the height differential between front and rear axle equal. Adjusted the camber variation accordingly as the technical specifications state.
- Changed the longitudinal pushrod point in order to transfer more load to the front of the car.
- Readjusted Caster so the angles keep being the same as baseline setup.

### 3.7.3. Custom setup results

The following graphics show the custom setup vs the base setup performing the same manoeuvres explained in section 3.5.2. A brief explanation of all of them will be made to compare the results.

#### 3.7.3.1. Roll motion comparison

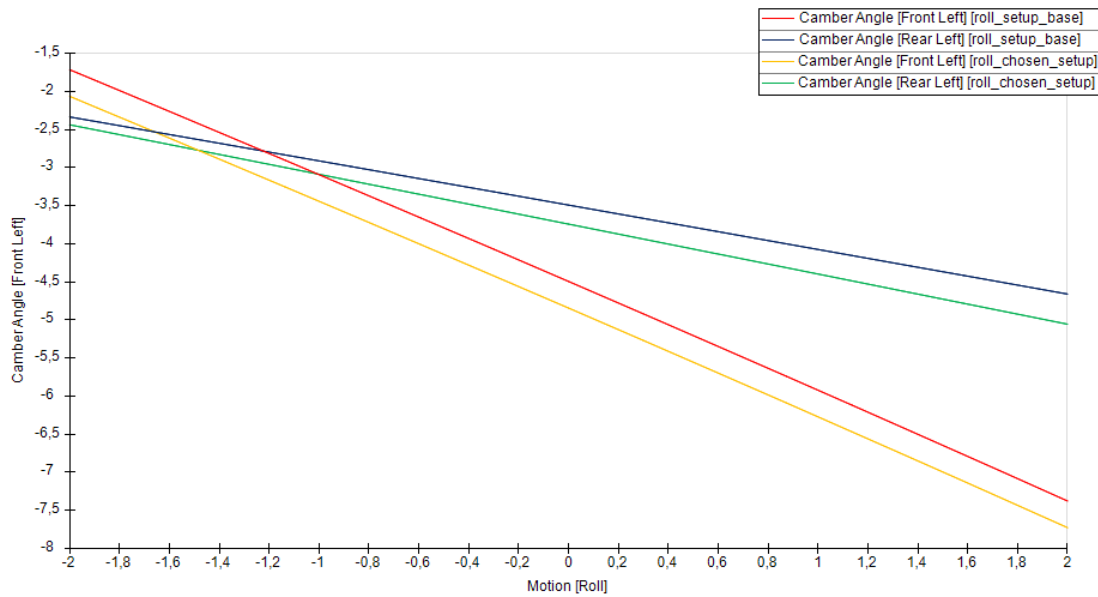


Figure 3.80: Camber angle vs Roll

It is observable that the chosen setup has gained more camber in both the front and rear axle respect the base setup. This enhanced performance during corner, as it ensures to have more camber thrust force during cornering events.

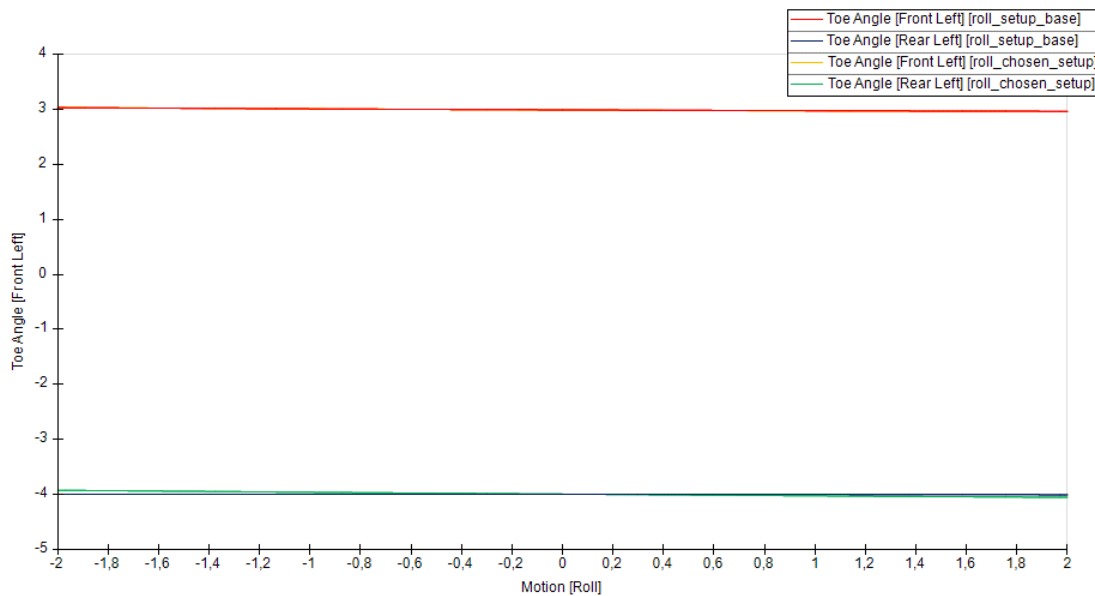


Figure 3.81: Toe angle vs Roll

In figure 3.88 it can be seen that the chosen setup retains the good properties the baseline setup has, as it has no toe angle variation during cornering events, thus helping to keep the behaviour of the car constant.

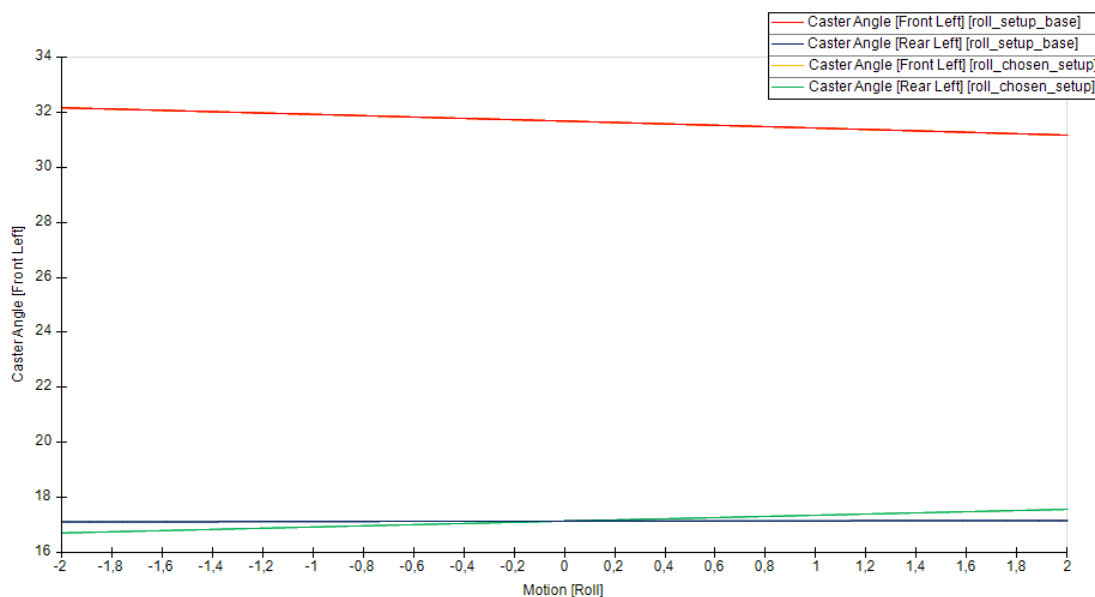


Figure 3.82: Caster angle vs Roll

Caster angle in the chosen setup is almost identical to the baseline setup, thus maintaining the force required at the wheel to perform a corner.

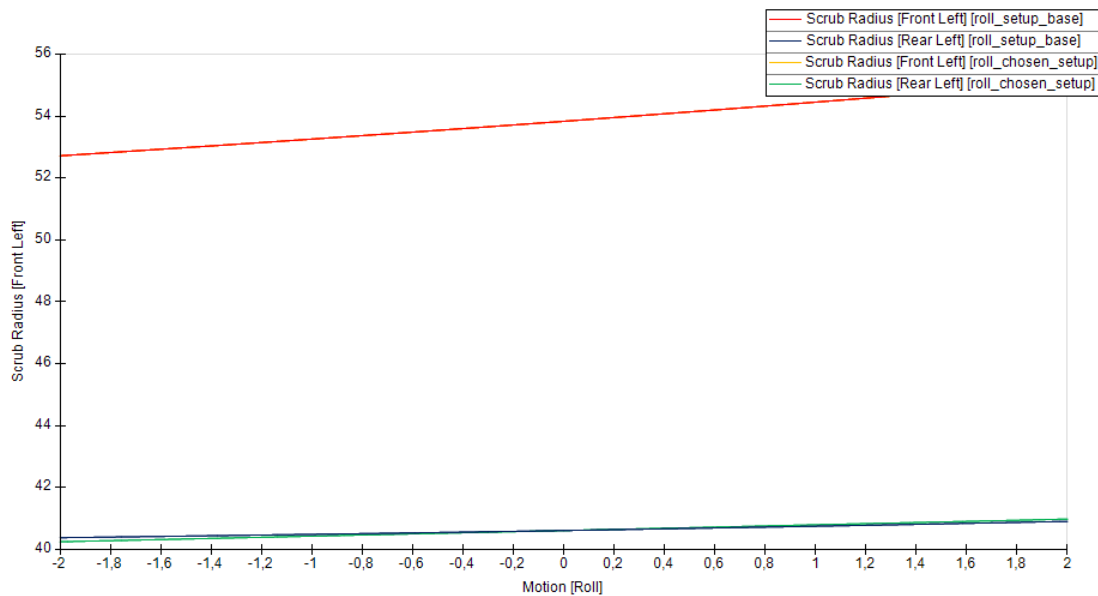


Figure 3.83: Scrub radius vs Roll

In figure 3.90, the scrub radius is almost identical to the baseline setup. Variations are subtle and tend to be linear, ensuring that the behaviour of the vehicle is constant during cornering events.

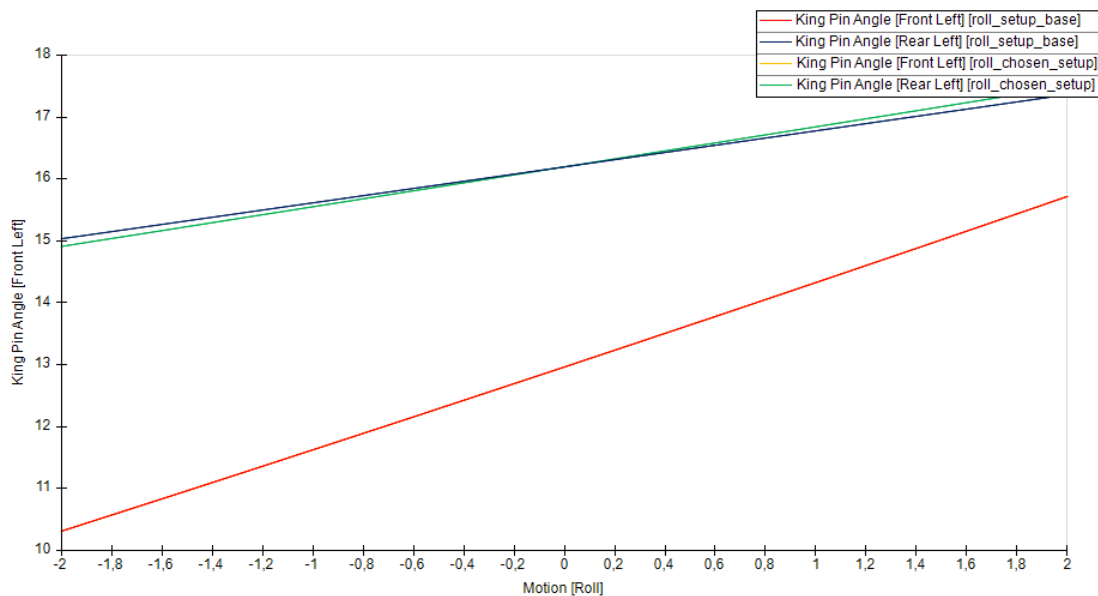


Figure 3.84: King pin angle vs Roll

The King Pin Angle is the same as the base setup as it has been corrected using the description provided by the manufacturer to keep it the same as the baseline setup.

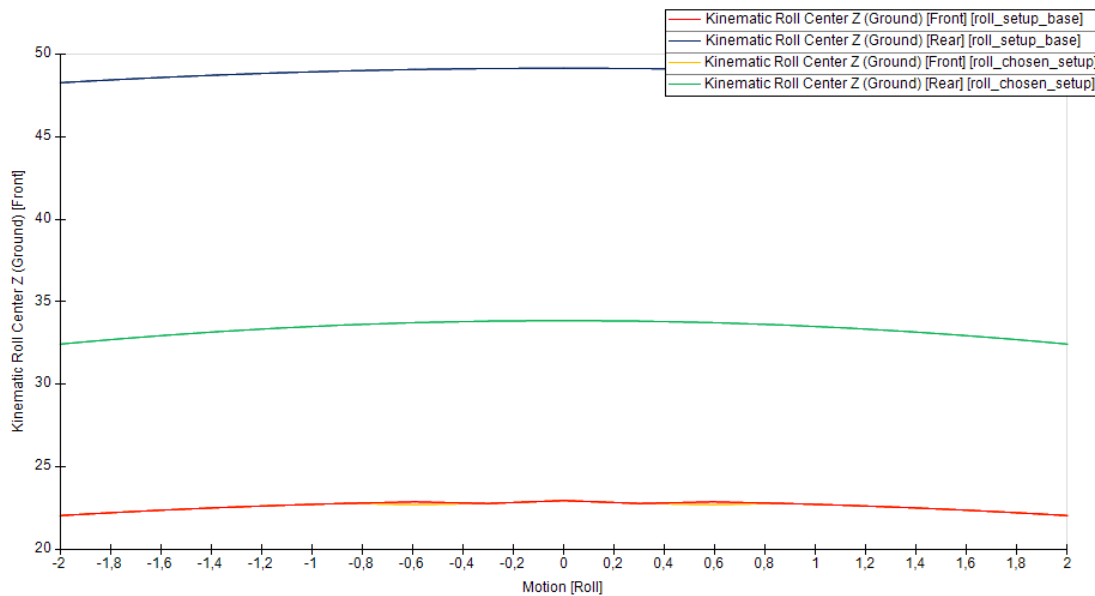


Figure 3.85: Kinematics roll center Z vs Roll

Here it is observable that the suspension pickup points chosen in the rear axle have made the rear roll center to go down, like the manufacturer stated in the user's guide. However, it is still above the front roll center during all the movement.

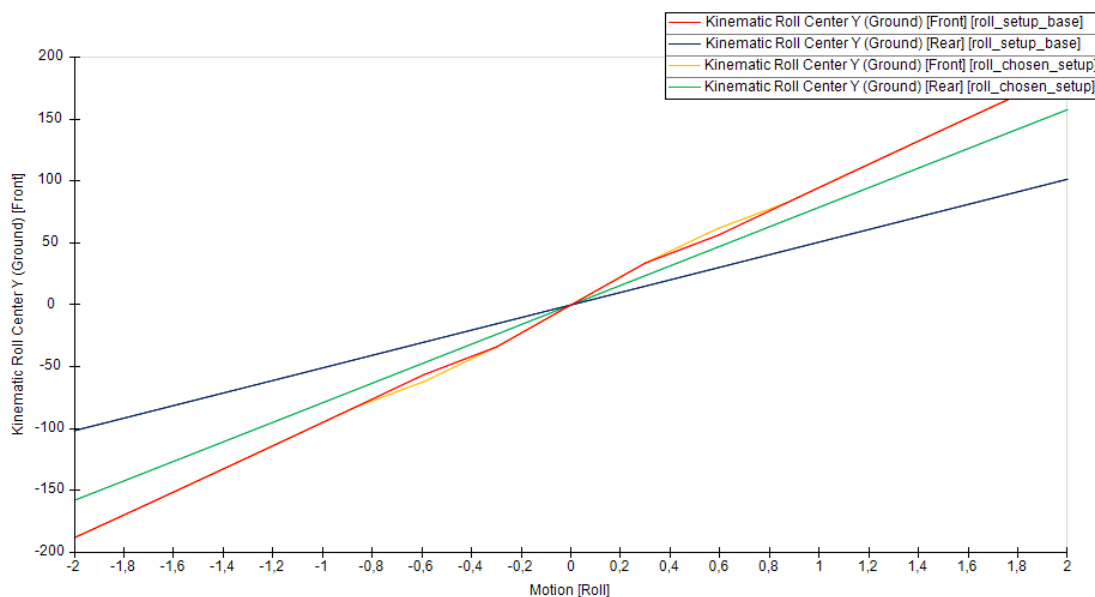


Figure 3.86: Kinematic roll center Y vs Roll

The roll center migrations are similar to the ones the base setup had, although the rear roll center has worsened a bit compared to the baseline value defined by the manufacturer.

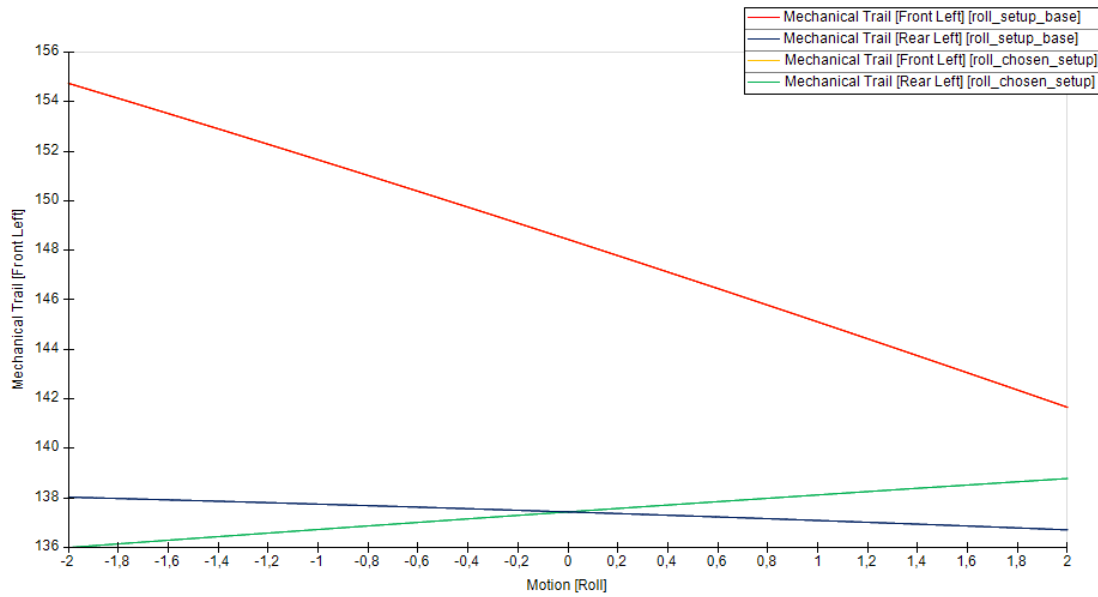


Figure 3.87: Mechanical Trail vs Roll

In figure 3.94, the noticeable change is the direction of the rear mechanical trail. Although similar to the value of the baseline, the configuration chosen for the rear suspension double a-arms changes the behaviour of the rear mechanical trail, as it is now opposite to the behaviour of the base setup.

### 3.7.3.2. Heave motion comparison

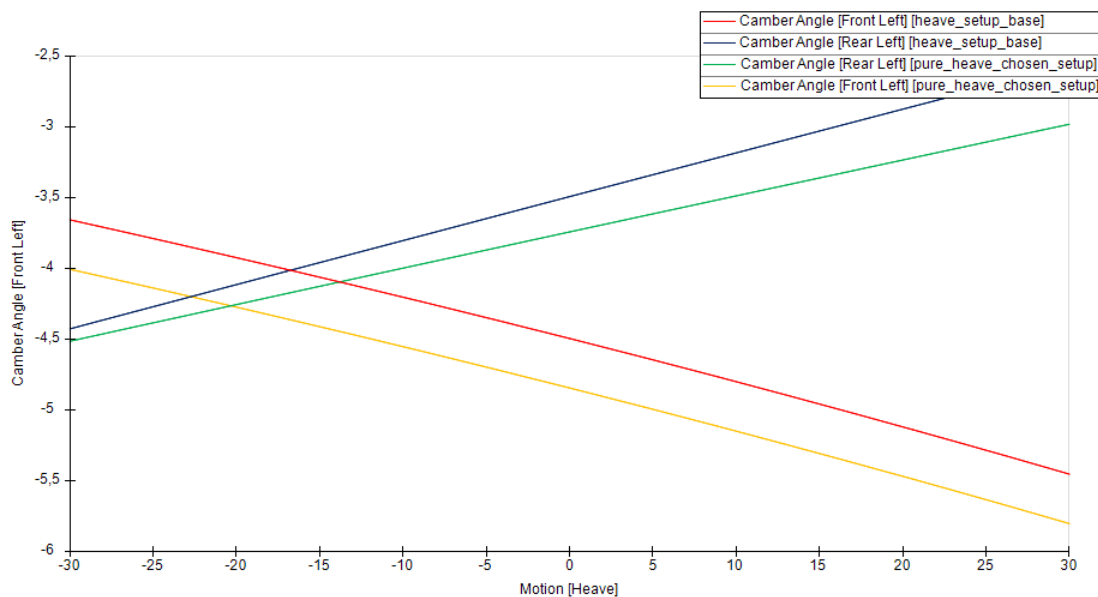


Figure 3.88: Camber angle vs Heave

Just like in the roll case, the chosen setup has gained a significant amount of camber respect to the base setup. Ensuring more grip for the wheels in a bump event.

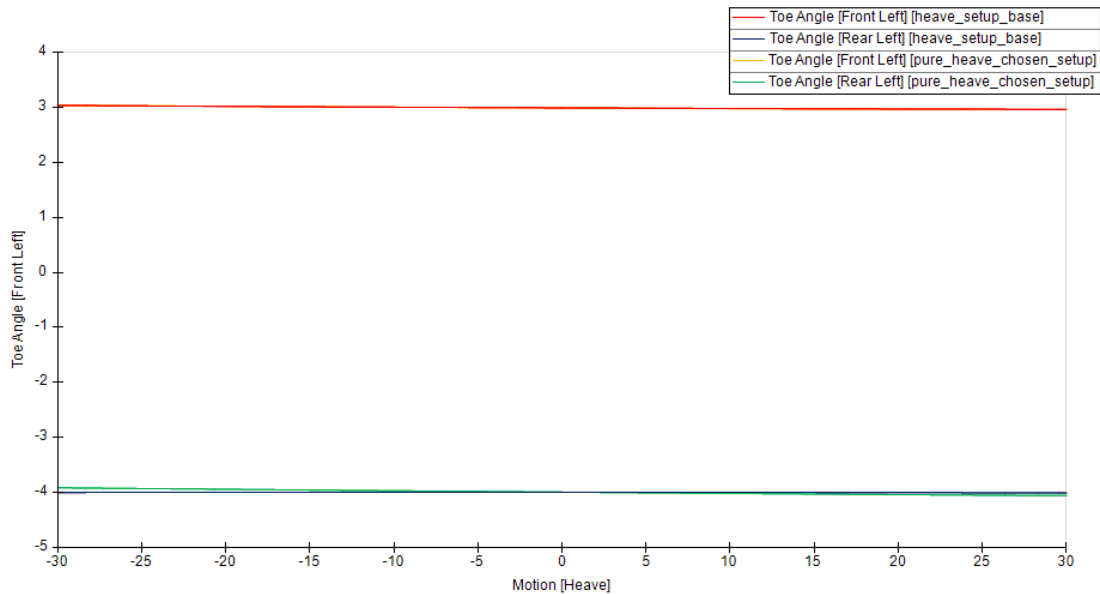


Figure 3.89: Toe angle vs Heave

The toe angle is the same as the baseline setup. Therefore ensuring that there are no bump steer effects and that the behaviour of the car is kept constant in bump events.

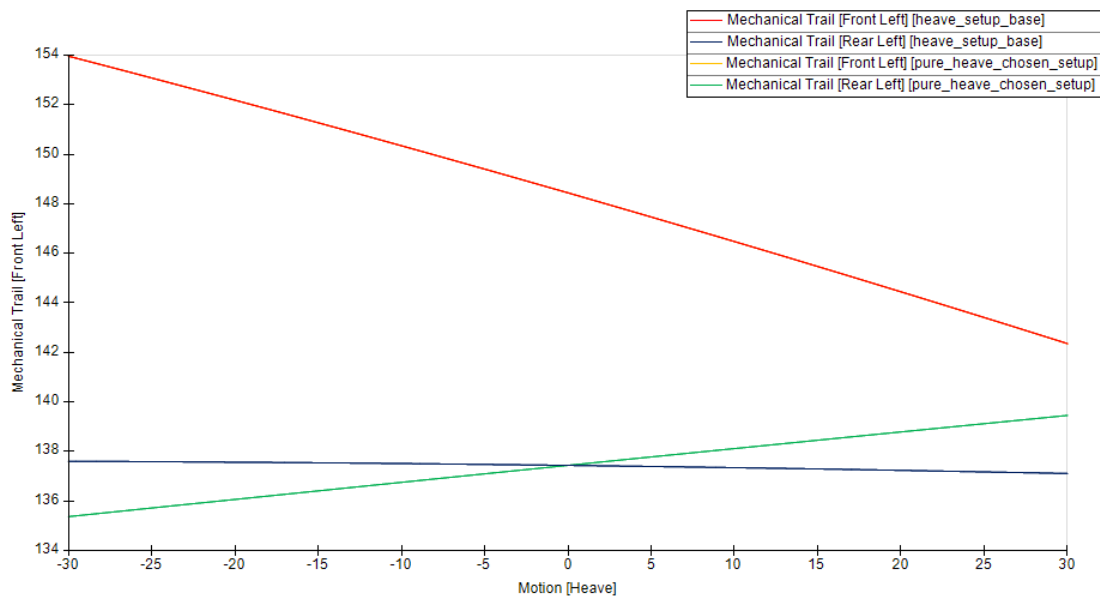


Figure 3.90: Mechanical Trail vs Heave

Just like in the roll motion case, the mechanical trail for the rear axle changed behaviour compared to the base setup. However, it retains mostly the same values as the previous baseline setup.



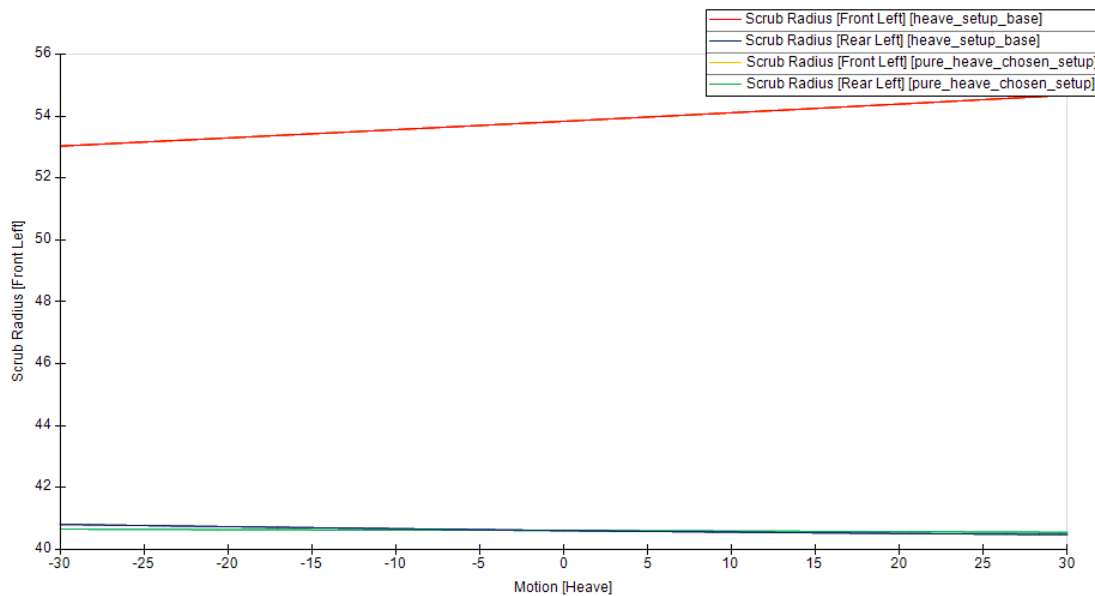


Figure 3.91: Scrub radius vs Heave

The scrub radius is exactly the same as the baseline setup, this ensuring a constant behaviour of the car during a bump event.

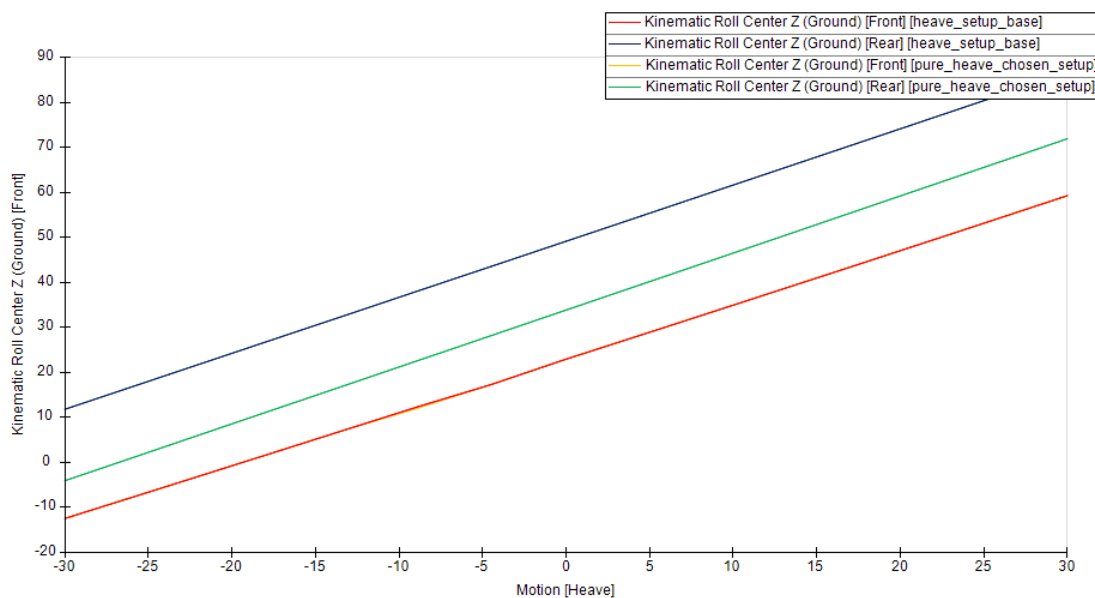


Figure 3.92: Kinematic roll center Z vs Heave

Just like it happened in the roll motion case, the chosen configuration for the rear axle's pickup points made its roll center to lose a bit of altitude. However it is still above the front roll center during all of the motion.

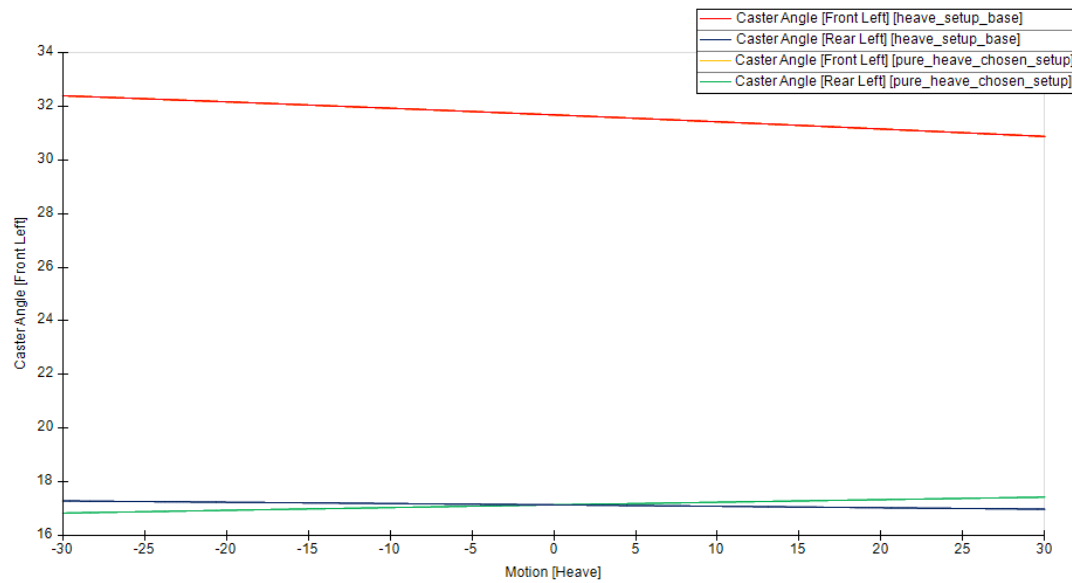


Figure 3.93: Caster angle vs Heave

The caster angle is almost identical to the one the baseline setup has. Therefore, it ensures that the car has a constant force at the wheel point during bump events because the variations are smoothly and mostly constant.

### 3.7.3.3. Steering motion comparison

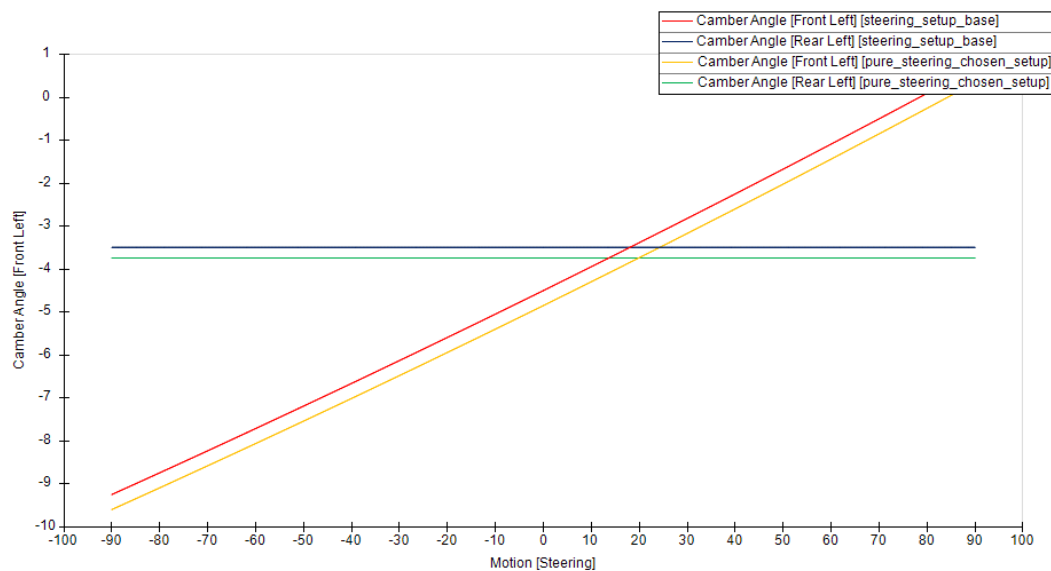


Figure 3.94: Camber angle vs Steering

Just like it happened in both of the previous movements, in figure 3.101 the custom setup allows

to assume more camber in both the front and the rear axle, ensuring to have more grip during all of the steering events of the car.

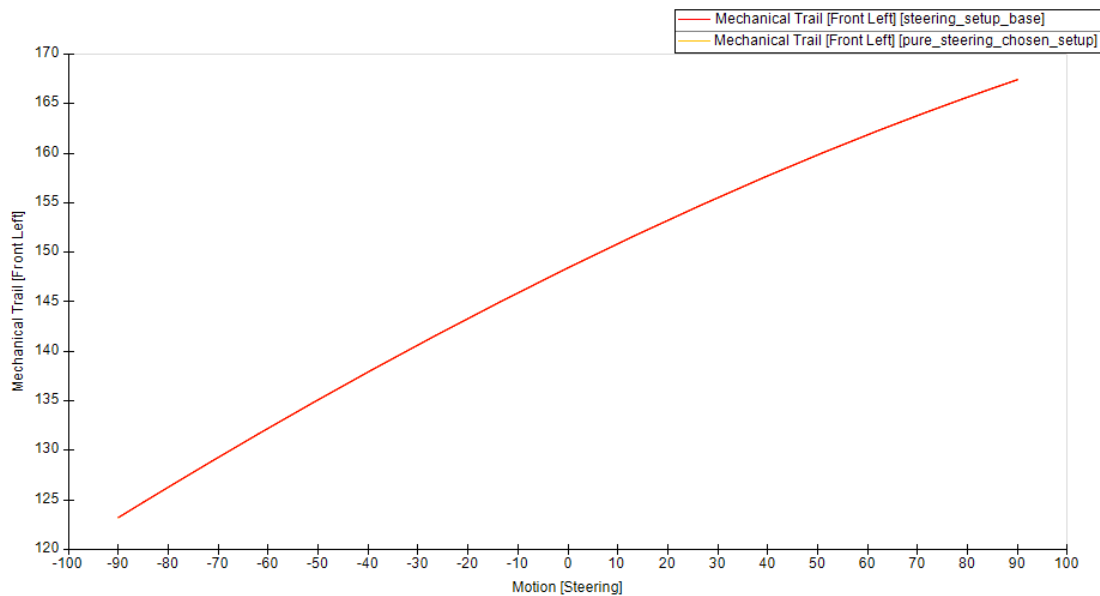


Figure 3.95: Mechanical Trail vs Steering

As the front axle's pickup points remain the same as the baseline setup, the mechanical trail is the same as the baseline configuration.

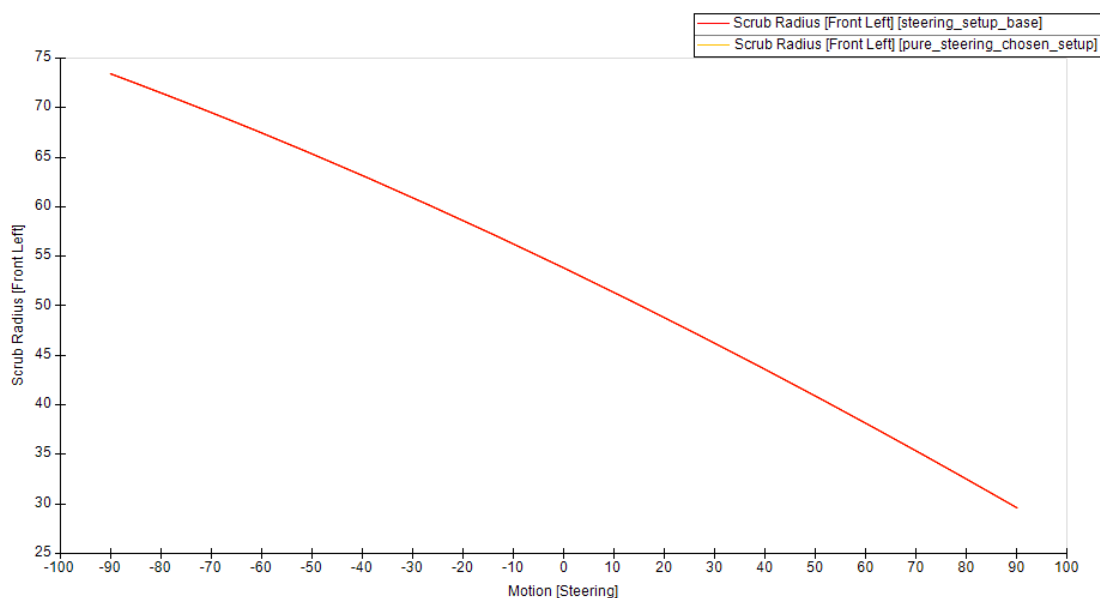


Figure 3.96: Scrub Radius vs Steering

The scrub radius in figure 3.103 remains the same as the baseline setup, as the steering's geometry has not been modified respect to the baseline configuration recommended by the

manufacturer.

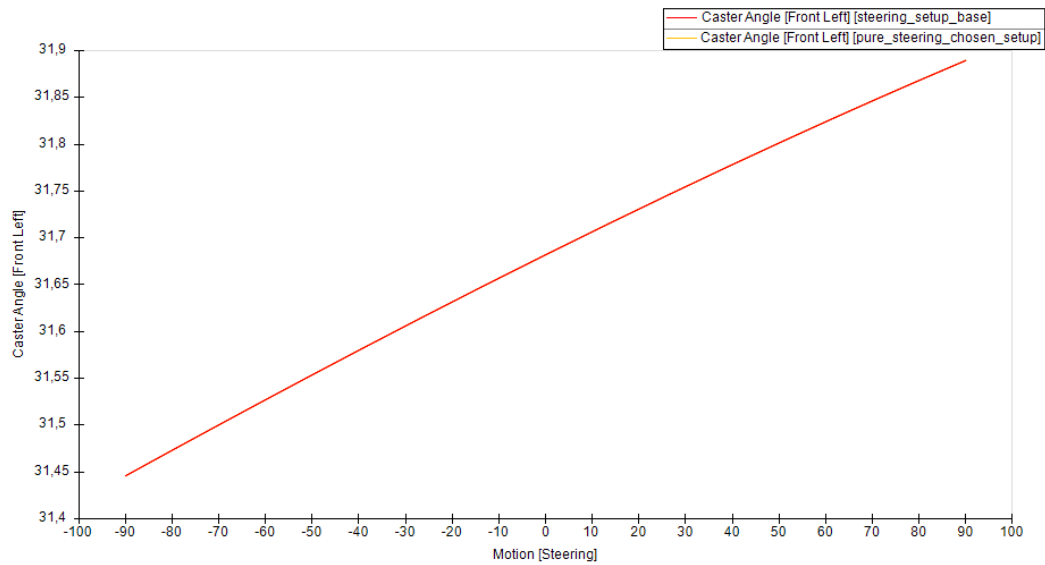


Figure 3.97: Caster angle vs Steering

### 3.7.3.4. Pitch motion comparison

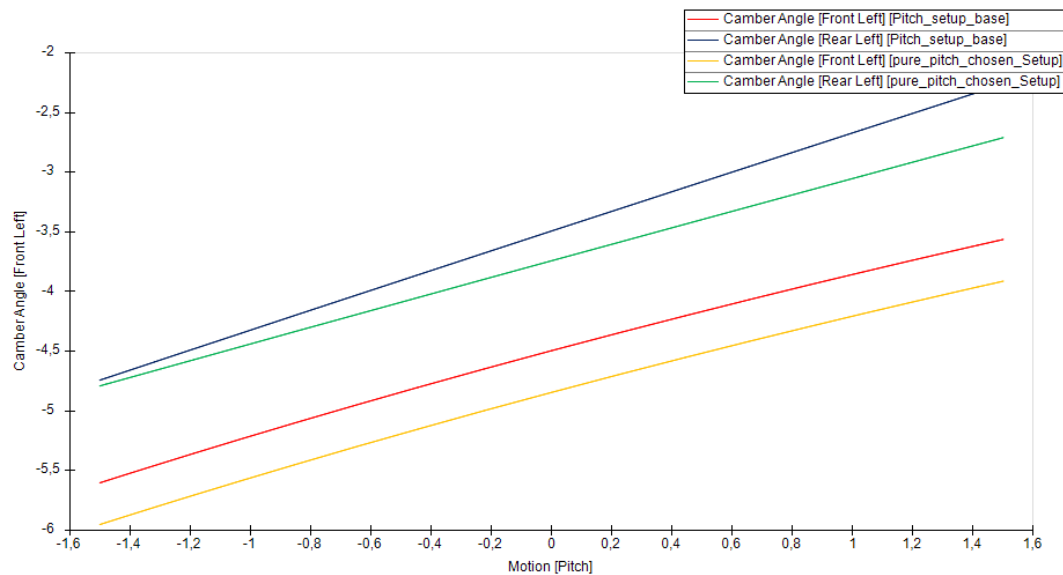


Figure 3.98: Camber angle vs Pitch

In figure 3.105, there has been an overall increase in camber angle for both axles, enhancing both grip during pitch events and the contact of the tyre with the surface.

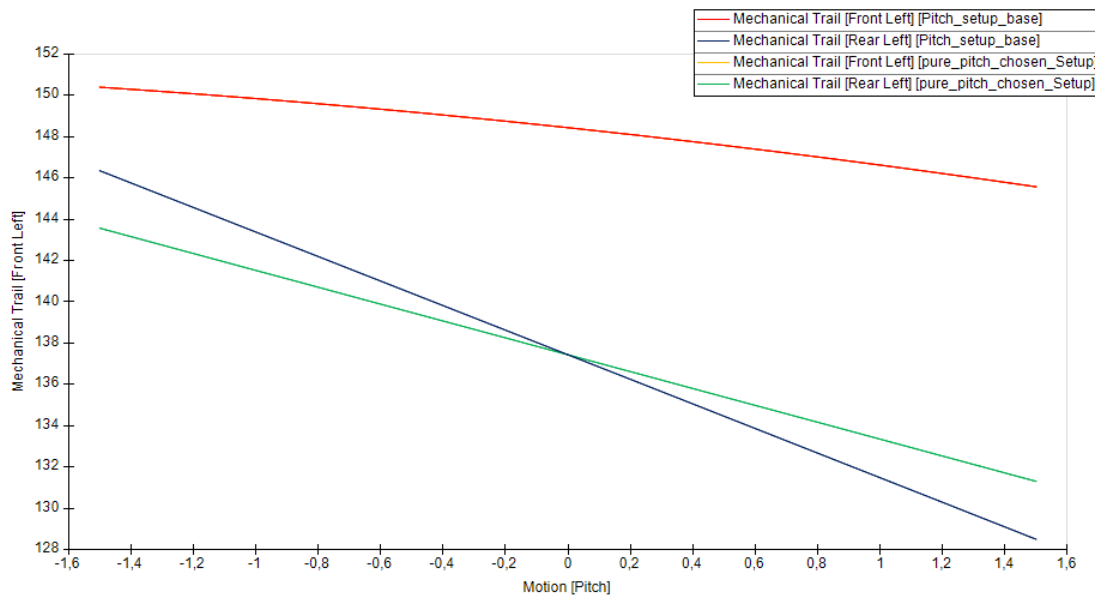


Figure 3.99: Mechanical Trail vs Pitch

The values remains pretty much similar to the baseline setup, with the rear mechanical trail value assuming slightly higher values on both ends of the movement, but not affecting the handling of the car.

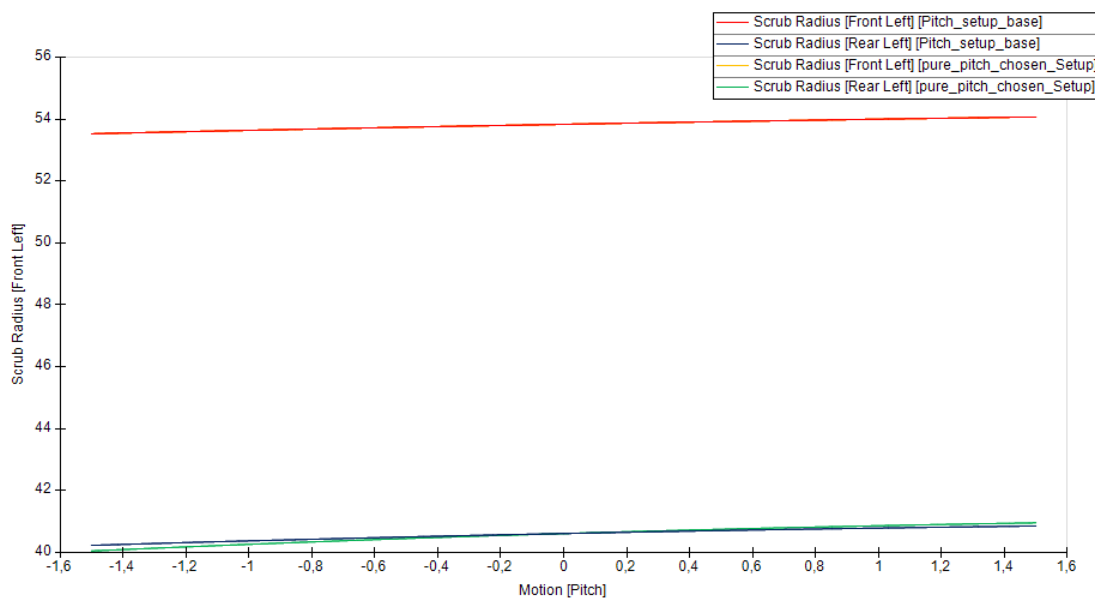


Figure 3.100: Scrub radius vs Pitch

The scrub radius is exactly the same in figure 3.107 as the baseline setup, this ensuring a constant behaviour of the car during a pitch motion.

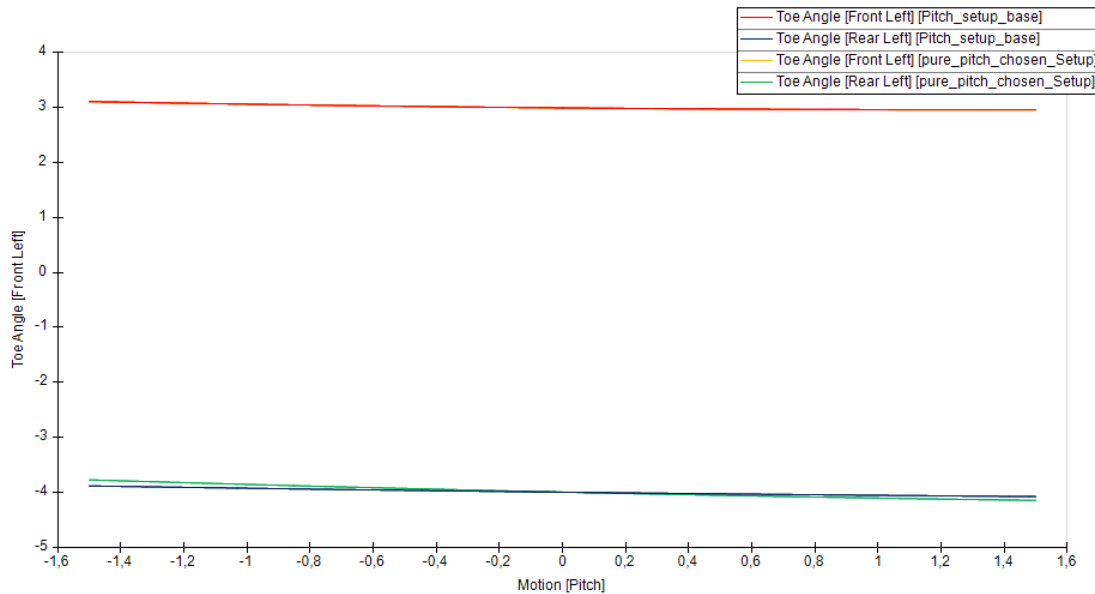


Figure 3.101: Toe angle vs Pitch

Just like in the previous motions, the toe values remain almost identical to the baseline setup, ensuring that the car does not change handling due to opening or closing the wheels during an acceleration or braking event.

### 3.7.3.5. Corner entry comparison

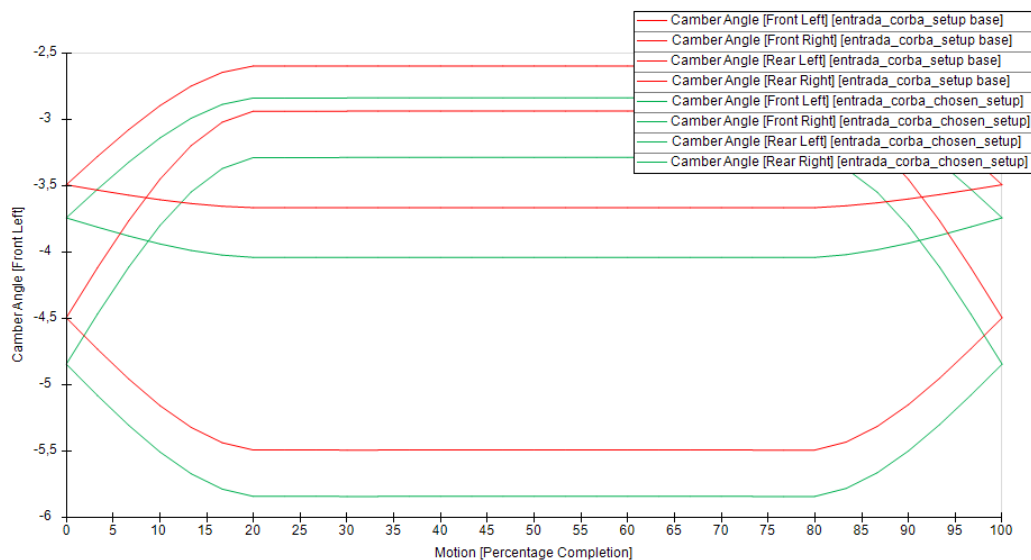


Figure 3.102: Camber angle vs motion completion in corner entry

In this advanced movement it can be seen that the camber angle in the overall movement is increased, allowing for a faster entry, consequent of having more camber thrust available.

### 3.7.3.6. Corner exit comparison

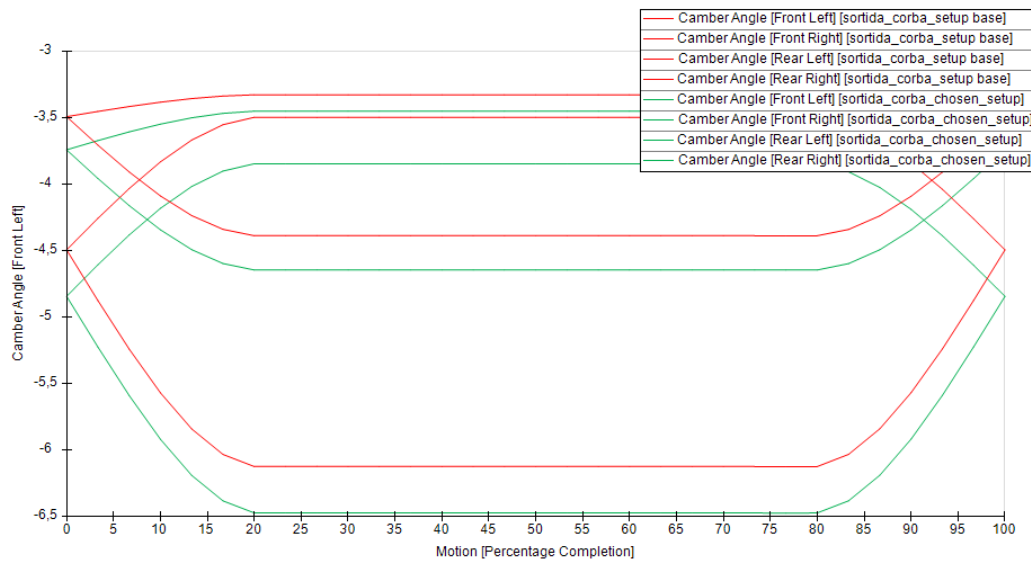


Figure 3.103: Camber angle vs motion completion in corner exit

Identically to the explanation of the advanced corner entry movement, in figure 3.110 the same analysis applies to this movement. There is more camber available, therefore the car is theoretically faster in an acceleration out of a corner.

### 3.7.4. Custom setup performance comparison

Using the Optimum Lap software defined in previous sections, the chosen setup is introduced and several analyses are performed in order to test if the setup defined in Optimum Kinematics related to faster lap times and increased performance on track.

Doing a simulation where fast laps are turned in the track trying to extract maximum performance, both of the configurations are analysed in the following table:

KPI Description	Circuit de Spa-Francorchamps	
Vehicle	Dallara F317_ Base_Setup	Dallara F317_ Chosen Setup
Lap time [s]	120,46	119,33
Percent in Corners [%]	74,49	74,34
Percent Accelerating [%]	49,49	45,27
Percent Braking [%]	32,48	35,77
Percent Coasting [%]	0,22	4,10
Percent 100% Throttle [%]	24,85	25,18
Percent TCS Enabled [%]	7,92	9,71
Lowest Speed [km/h]	66,34	69,95
Highest Speed [km/h]	229,95	229,95
Average Speed [km/h]	214,80	216,25
Energy Spent [kJ]	18221,47	17412,41
Fuel Consumption [kg]	1,54	1,48
Gear Shifts [-]	3984,00	4212,00
Maximum Lateral Acceleration [m/s <sup>2</sup> ]	-50,05	-52,09
Maximum Longitudinal Acceleration [m/s <sup>2</sup> ]	11,05	12,15
Maximum Longitudinal Deceleration [m/s <sup>2</sup> ]	-55,58	-60,25

Table 3.13: Comparison table of the base setup vs the custom setup

It is observable that the base setup laps the track in 2:00:460, whereas the chosen setup laps it in 1:59:330. Therefore, there has been an improvement of 1,16 seconds per lap.

It is also observable that the chosen suspension setup fulfils its purpose further enhancing the contact of the tire with the surface of the track, being able to assume more lateral and longitudinal acceleration. To further exemplify this, 2 more graphs were plotted in order to show the lateral acceleration and speed through the corners of the track versus the elapsed distance for both the base setup and the chosen setup.



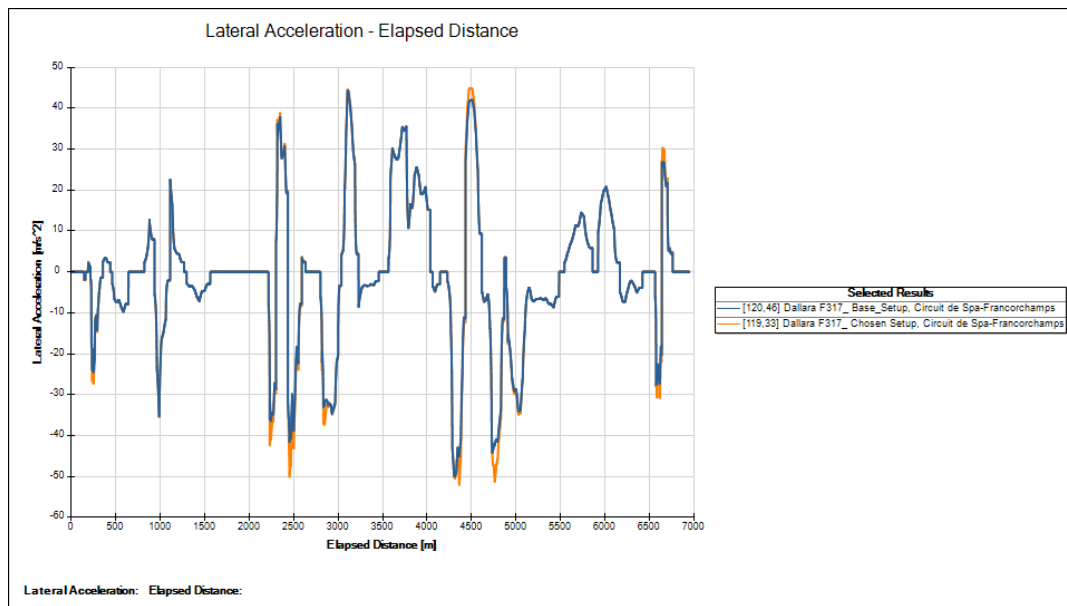


Figure 3.105: Lateral acceleration vs Elapsed distance

It is observable that during medium-fast paced corners from Les Combes to the Stavelot section of the track the chosen setup is able to achieve bigger lateral acceleration values

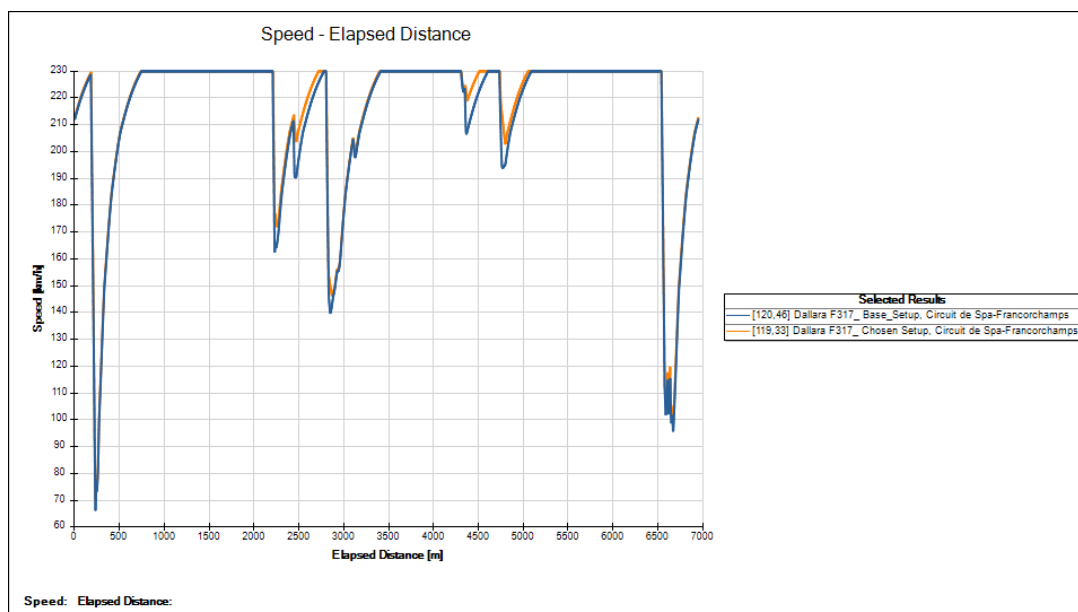


Figure 3.106: Speed vs Elapsed distance

As a consequence of the chosen setup for having better tire contact with the surface, the speed through the middle section of the track is enhanced noticeably, contributing to the lap time difference shown in the table.

### 3.8. Project timeline

Table 3.14 shows the project timeline with the aim of organizing and showing the time dedicated to each of the tasks needed to develop the project. The duration of the project is fixed to 3 months.

TASKS	WEEKS															
	18-24 SEP	25-1 OCT	2-8 OCT	9-15 OCT	16-22 OCT	23-29 OCT	30-5 NOV	6-12 NOV	13-19 NOV	20-26 NOV	27-3 DIC	4-10 DIC	11-17 DIC	18-24 DIC	25-31 DIC	
Information and Organization of the project																
Information filtering																
State of the art - Champioship categories and background																
State of the art - Manufacturer and the Dallara F-317																
State of the art - Suspensions of the Dallara F-317																
Design analysis of the suspension system																
Analysis of the initial configuration of the suspension system																
Kinematic analysis of the initial configuration of the suspension system																
Lap time analysis of the initial configuration of the suspension system																
Kineramtic onfiguration of the configuration of the suspension system																
Lap time analysis of the proposed configuration of the suspension system																
Lap time performance comparison																
Conclusions																
Bibliography and adequation of the format																

Table 3.14: Project timeline

### 3.9. Environmental impact

This project is developed entirely in a virtual environment, therefore there is no notable environmental impact. The electric consumption of the electronic equipment has been included into the price/hour category of Table 3.15 in the section of the budget analysis.

### 3.10. Budget analysis

The following section covers the economical cost of the realization of the project from the definition of the idea of the project until the ending of the project's writing.

Table 3.15 is based on table 3.14 as it covers the hours dedicated to each of the tasks detailed in the project timeline. The cost per hour for the engineering hours dedicated to each task is estimated and defined by own elaboration making a ponderation of different values found in this article. [15] The degradation of electronic equipment and cost of the electricity for the equipment is included in the engineering hours.

ENGINEERING HOURS			
Concept	Hours	Price/hour (€/h)	Total cost
Information and Organization of the project	20	15	300
Information filtering	8	15	120
State of the art - Championship categories and background	8	15	120
State of the art - Manufacturer and the Dallara F-317	8	15	120
State of the art - Suspensions of the Dallara F-317	8	15	120
Design analysis of the suspension system	15	15	225
Analysis of the initial configuration of the suspension system	15	15	225
Kinematic analysis of the initial configuration of the suspension system	15	15	225
Lap time analysis of the initial configuration of the suspension system	20	15	300
Kinematic configuration proposed for the suspension system	60	15	900
Lap time analysis of the proposed configuration of the suspension system	20	15	300
Lap time performance comparison	15	15	225
Conclusions	8	15	120
Bibliography and adequation of the format	20	15	300
Reunions	20	15	300
Field Work	15	15	225
TOTAL	275		4125
SOFTWARE			
Concept	Price (€)		
Optimum Kinematics	2.116,37 €		
Optimum Dynamics	3.177,06 €		
TOTAL	5.293,43 €		
TOTAL COST OF THE PROJECT		9.418,43 €	

Table 3.15: Budget analysis of the project

## Conclusions

In accordance to the established objectives of the project, the following conclusions were achieved:

1. In the project, a general overview of the state of the art of the suspension systems of a vehicle is exposed, the beginnings of the Formula 3 championship and the correspondent competition regulations.
2. The technical characteristics of the Dallara F-317 are detailed, serving as the starting point for the study of the project.
3. The suspension system of the Dallara F-317 is analysed. It is modelled and simulated using the default configuration established by the manufacturer. Other configuration options that produce adequate performances of the car are compared with the starting point configuration.
4. Based on the results of the simulation done with the Optimum Kinematics software, new configuration of the suspension system for the front and rear axle is proposed to compete in the specific track of Spa Francorchamps.
5. After the simulations done in Optimum Lap, it is validated that the new proposed configuration for the suspension system offers better results than the base configuration in a virtual lap time simulation in the Spa Francorchamps track.

## Mentions

The following persons and organizations helped to made this study possible:

The UPC ETSEIB School; for offering me quality knowledge about engineering, developing my skills to research information and understanding of technical engineering documentation and for offering the ETSEIB Motorsport program in its program.

The ETSEIB Motorsport team; for being able to be part of the team for three years. Learning all the field related information, techniques and tricks to develop a racing car.

Enrique Ernesto Zayas Figueras; for tutoring my project, for his continuous support and assistance and for offering me the global vision needed to commit to each of the sections of the study without losing focus of the global objective of the project.

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